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REVIEW OF ENGINEERING PROGRESS ABROAD

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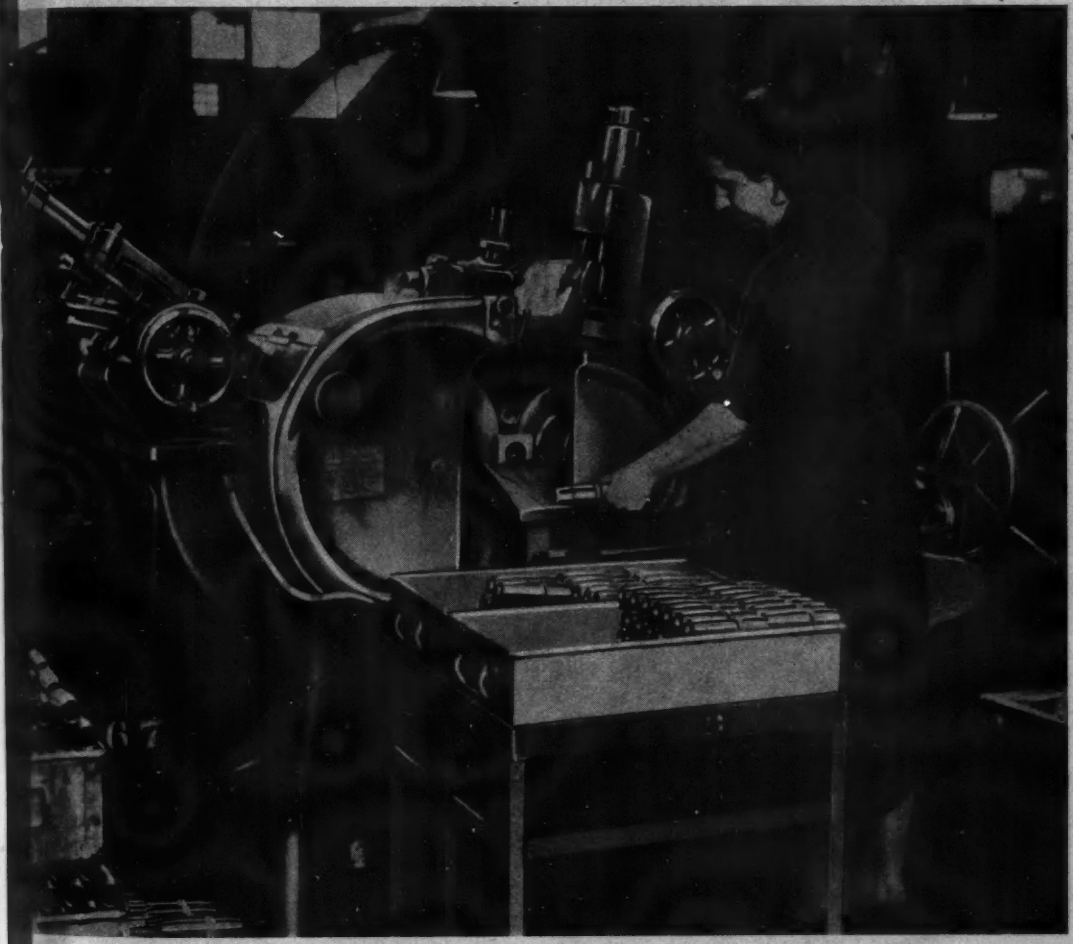

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CONTENTS

	page
The Aerodynamic Turbine *versus* Steam and Gas Turbines *"The Engineers' Digest," Vol. 5, No. 2, p. 34.*	34
Characteristics of Black Coatings for Non-Ferrous Metals *"The Engineers' Digest," Vol. 5, No. 2, p. 57.*	57
Development of Cast Crankshafts *"The Engineers' Digest," Vol. 5, No. 2, p. 58.*	58
Double-Exposure Radiography *"The Engineers' Digest," Vol. 5, No. 2, p. 58.*	58
The Ericsson Automatic Telephone System *"The Engineers' Digest," Vol. 5, No. 2, p. 48.*	48
The French Gas Producer Industry in 1943 *"The Engineers' Digest," Vol. 5, No. 2, p. 53.*	53
The Laurette Oil Filter *"The Engineers' Digest," Vol. 5, No. 2, p. 49.*	49
New Oil-Blast Breaker for Medium Voltage *"The Engineers' Digest," Vol. 5, No. 2, p. 46.*	46
Operating Economy of Pneumatic and Steam Forging Hammers *"The Engineers' Digest," Vol. 5, No. 2, p. 31.*	31
Seven-Spindle Horizontal Automatic for Machining Large Work-Pieces *"The Engineers' Digest," Vol. 5, No. 2, p. 29.*	29
Several Problems on Torsional Vibration in Internal Combustion Engines *"The Engineers' Digest," Vol. 5, No. 2, p. 42.*	42
The Soviet W 2 Tank Diesel *"The Engineers' Digest," Vol. 5, No. 2, p. 39.*	39
Strength of Arc Welded Seams *"The Engineers' Digest," Vol. 5, No. 2, p. 51.*	51
Testing of Motor and Generator Armatures by Means of Cathode Ray Tubes *"The Engineers' Digest," Vol. 5, No. 2, p. 55.*	55

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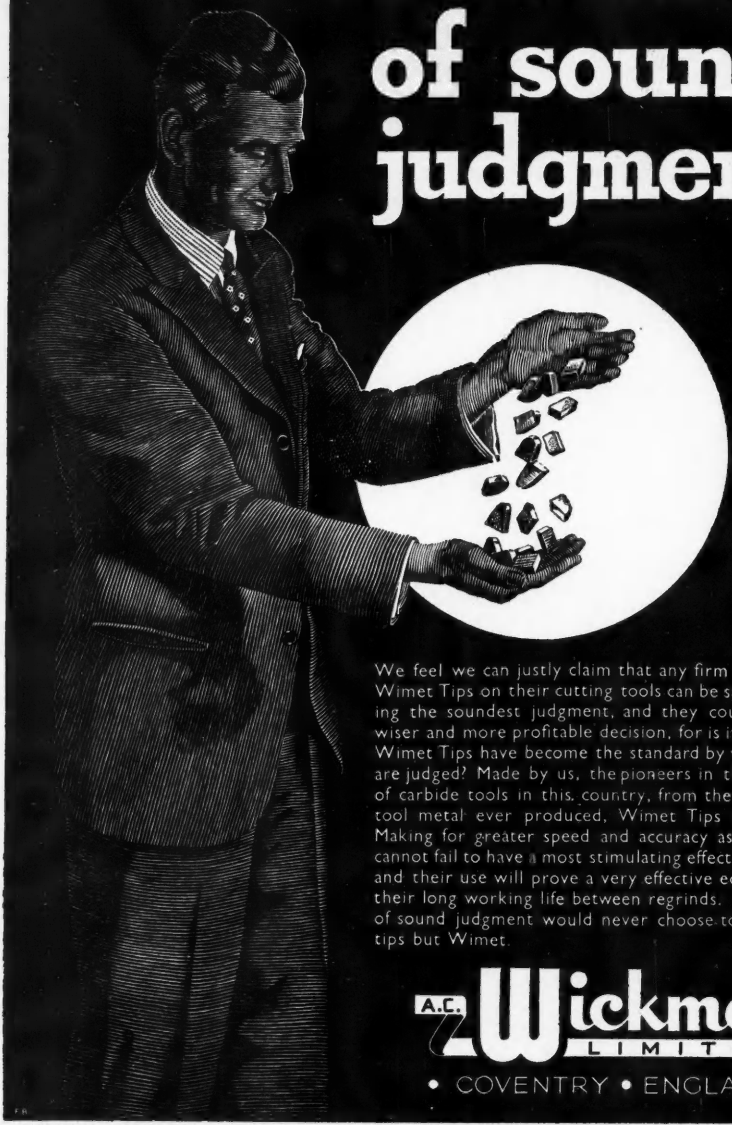

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SEVEN-SPINDLE HORIZONTAL AUTOMATIC FOR MACHINING LARGE WORK-PIECES

By E. DORNHÖFER. (From *Zeitschrift des VDI.*, Vol. 87, No. 31/32, Aug. 7, 1943, pp. 489-492).

CONSIDERATIONS of man-power and the necessity of bringing about a decrease in production costs motivated the carrying out of an investigation into the suitability of a multi-spindle automatic for the machining of large work-pieces which are supplied in a rough-machined state by a forge. Up to the present, roughing operations have been carried out with boring mills, while final machining has been done in ordinary lathes. Because of the size and unusual shape of the work-pieces, a multi-spindle automatic of the usual type could not be used, and the employment of a vertical automatic had to be ruled out because of the difficulty of removing the chips from the pot-like work-pieces.

These considerations led to the construction of a horizontal machine with seven radially arranged spindles, in which six tool slides are embodied, the seventh headstock and spindle being used for chucking operations (Figs. 1-3). The spindles are carried in seven identical headstocks fixed on a round rotatable table, the latter in turn resting on a base bolted to the foundation. Each tool slide likewise forms, with its bed and drive, an integral unit bolted to the foundation. With the latter thus serving as the base for the assembly, care must be taken to give it adequate strength. Each spindle is driven by an individual motor, the electric power supply being effected over slip-rings arranged at the centre of the machine.

Normally the table stands still in a locked position while the tool slides are brought into operating position with accelerated speed, their individual movements being independent of each other. When the operating position is reached, tool feed is carried out automatically. At the end of their stroke—that is, after termination of the working period—the slides are reversed automatically by means of fixed stops ; and when all slides have returned to their initial position, the centre table is unlocked by a hydraulic device and moved on to the next stop, when it is locked again and the operating cycle repeated. While the table moves on, rotation of the spindle coming into chucking position is stopped automatically and the spindle moving into the first working position is started automatically. Since chucking is carried out while the six working spindles are in operation, no time is lost in handling the work-pieces which move stage by stage through the working cycle.

This seven-spindle machine, which had to be constructed in a comparatively short time, was found to meet all expectations. Care had, of course, to be taken to arrange for the supply of the rough pieces in suitable condition ; to provide the required tools in sufficient number, and to reduce the times for tool setting to a minimum. Since the machine reduces machining time to one fourteenth of that previously required, it performs the work of 14 boring mills. As one operator is able to attend to the machine, the work of twelve, and later of even thirteen, hands is saved. It must be taken into consideration, however, that the entire time of one tool

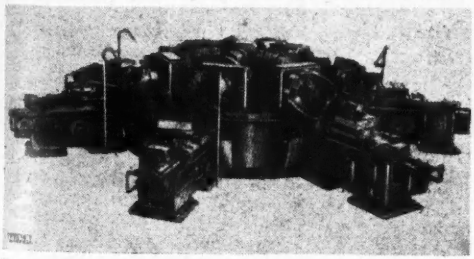
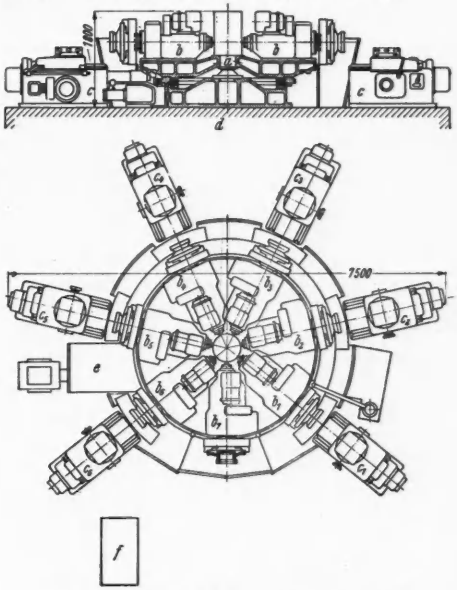


Fig. 1. **Seven-spindle horizontal automatic for machining large work-pieces.**



Figs. 2 and 3. **Arrangement of components.**
(a) Turn-table, (b) b_1-b_7 Headstocks, (c) c, c_1-c_6 Tool slides, (d) Foundation (also serves as a bed plate), (e) Operating mechanism, (f) Operating board.

setter is nearly required to tend two machines of this type. Based on an annual operating time of 2,500 hours, the total capital plus operating cost (excluding tools and overhead) of the machine is 13.35 RMk. per hour as compared with 2.92 RMk. for a single boring mill. With a certain production rate, the cost of the machine can be written off in 10 months, with the machining cost per piece reduced by two-third.

The often-voiced objection that such results can be achieved only where large numbers of pieces are concerned, only applies within certain limits. It is true enough that multi-spindle machines of special types involve considerable setting-up times ; but these times can be considerably reduced by adequate design of tool-holders and by keeping tools in readiness. Additional time can be saved by providing in the machine design means for quick and simple adjustment of travel and feed rates. In the machine referred to, travels are set by means of simple adjustment of the stops ; while feed rates are infinitely variable. Thus, the specific setting-up time chargeable per piece machined is largely a function of the machining time per piece. Since in the case considered, the machining time of the heavy and unwieldy work-pieces is long, the influence of the setting-up time is small, and production series may be as little as 25-50 pieces.

The machine cost per piece on the basis of a machining time of 60 min. per piece, and as influenced by the number of pieces produced, is charted in Fig. 4. Here it is seen that even with a production of three pieces, the production cost with the new machine equals that obtaining with a boring mill ; while for 100 pieces, the cost is reduced to 32 per cent of that hitherto obtaining. This figure cannot, however, be considered as generally applicable, since it is due to certain special circumstances. But to judge from the experience gained so far, the new machine may be expected to have a potential

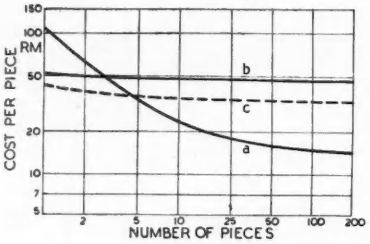


Fig. 4. **Comparative production cost per piece with**
(a) Automatic, (b) Boring mill, (c) Capstan lathe; Operating time per piece: 60 minutes.

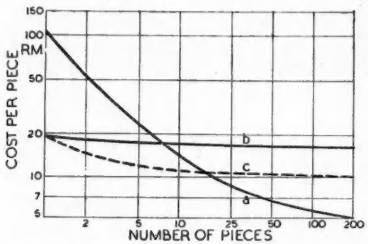


Fig. 5. **Comparative production cost per piece with**
(a) Automatic, (b) Boring mill, (c) Capstan lathe; Operating time per piece: 20 minutes.

production capacity approximating to seven times that of a heavy capstan lathe. Comparing relative production costs on the basis of 20 minutes operating time per piece machined, equal results are obtained with the new machine and with the capstan lathe when producing a series of 17 pieces with the new machine (Fig. 5). According to the same graph, for a series of 100 pieces, the production cost of the new machine is only 57 per cent of that recorded for the capstan lathe and 35 per cent as compared with the boring mill.

Owing to the high strength of 110 kg. per sq. mm. of the material to be machined, especially low feed rates were called for in order to secure a reasonable tool life, and small feed rates down to 1 mm. per minute with a feed force of up to 4,000 kg. had to be provided. Furthermore, reversing movements had to be initiated by fixed stops. It was felt that these requirements could not be safely met with in the usual type of hydraulic feed drive and a new type of feed mechanism was therefore developed. In this design (Fig. 6) the spindle in the tool slide (tool carriage) is driven by an oil-hydraulic drive and a worm gear when feeding, and through a spur gear when accelerated movement is required. The necessary clutches are of the multi-friction-disc type operated by a power piston. The hydraulic power unit, which is designed for constant torque and constant input of pressure oil, is supplied from an oil pressure pump of infinitely variable output and is driven by an electric motor via a reversing valve which allows the reversal of direction of rotation of the power unit. The speed of the latter can be varied within a certain range by adjusting the rate of oil supply from the pump.

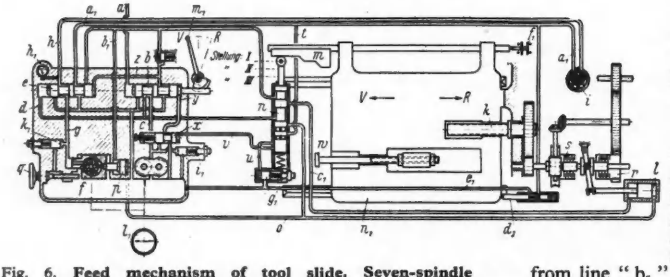


Fig. 6. **Feed mechanism of tool slide. Seven-spindle automatic.**
(a) Pressure oil pipe, (b) Auxiliary pilot valve, (c) (d) Oil piping to main pilot valve, (f) Infinitely variable pump, (g) (h) Oil pipes, (j) Power unit, (k) Spindle of tool slide, (l) Power piston, (m) Stop, (n) Pilot valve, (o) Oil pipe, (q) Control handwheel, (r) Clutch for accelerated slide drive, (s) Clutch for feed drive of tool slide, (t) (u) (v) Oil piping, (w) Stop, (x) Pilot valve, (y) (z) (a_1), (b_1), (c_1) Oil piping, (d_1) Shutdown valve, (e_1) Pipe, (f_1) Electric contact (g_1) Valve, (h_1) Pressure gauge, (i_1) Relief valve for pump f, (k_1) Relief valve for oil pressure system, (l_1) Electric motor, (m_1) Control lever, (n_1) Tool slide.

Pilot pressure oil is supplied by a separate pump of the constant volume type which is coupled direct to the aforementioned main pressure oil pump. An auxiliary pilot valve and a valve for operating the friction-disc clutches for the ordinary feed and accelerated feed drives, as well as a shutdown valve are also provided. The table is locked by the action of hydraulic pump "f" (Fig. 6) (supplying pressure oil for movement and for locking the rotatable table of the machine) and pressure is transmitted through line "a" which brings the auxiliary pilot valve "b" into position "V" (forward movement). Pressure oil can now flow from pilot pressure pump "f" to the main pilot valve "e" via "c" and "d." The variable pump "f" at this moment adjusted to maximum delivery, supplies pressure oil via the lines "g" and "h" to the power unit "i," setting it in motion. As the piston of the power cylinder "l" is in the extreme r.h. position, this motion is transmitted via the spur gear drive and the coupling to spindle "k," thus advancing the slide at accelerated speed. When stop "m" on the slide moves pilot valve "n" into position II., pressure oil passes through line "o" to the r.h. side of piston "p" and forces the impeller of the variable pump "f" against a spindle which is adjusted by the control hand-wheel "g." At the same time oil passes to the r.h. side of piston "l" and moves it to the left, disengaging the coupling "r" for accelerated movement and engaging the coupling "s" for ordinary feed speed. The feed rate is then controlled by means of hand-wheel "g."

Shortly before the tool slide has reached the final position to which its stop "w" is set, it engages another stop "m," shown in Fig. 7, which moves the piston of the pilot valve "n" into position III., thereby connecting line "t" with lines "u" and "v." When the slide subsequently reaches stop "w," the increased torque of the power unit "i" increases the oil pressure in line "h," this pressure impulse being transmitted through lines "t" and "v" to pilot valve "x" set at a fixed pressure. The pressure impulse moves the piston of valve "x" to the left, thus opening connection "y" and admitting pressure oil on the r.h. side of the pilot valve "b," which is correspondingly moved to the left, thus permitting the flow of pressure oil from connection "c" through line "z" to the r.h. side of the main pilot valve "e." The piston of the latter, therefore, moves to the left and now obtains pressure oil through the line "a_1" instead of through line "h." The power unit therefore begins to rotate in the opposite direction. At the same time, the piston "p" receives pressure oil at the left from line "b_1" which causes the pump to give maximum delivery. The tool slide now moves back, but only slowly, because of the intervention of inertia effects. But as soon as stop "m" has released position III., line "c_1" receives oil from line "b_1," which oil acts upon the l.h. side of the coupling, thus disengaging the normal-speed clutch and engaging the accelerated-speed clutch. The slide now moves at accelerated speed until it contacts with the valve "d," and cuts the

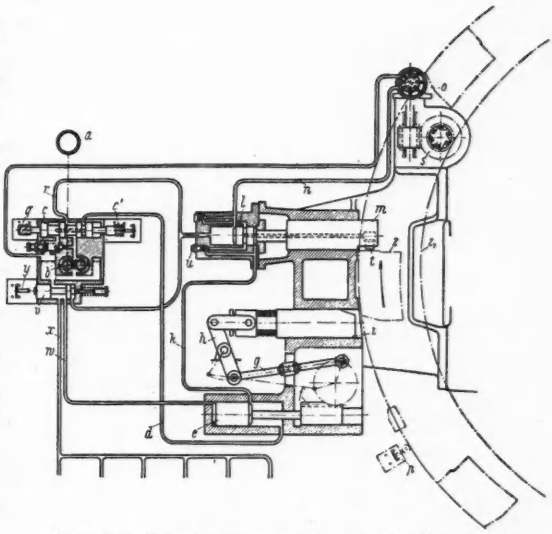


Fig. 7. **Operating and locking mechanism of turn-table.**
(a) Electric motor, (b) Gear type pump, (c) (c_1) Pilot valves, (d) Oil pipe, (e) Piston, (f) Gear rack, (g) Push rod, (h) Lever, (i) Bolt, (k) Oil pipe, (l) Piston, (m) Stop, (n) Oil pipe, (o) Power unit, (p) Electric contact, (q) Solenoid, (r) Oil pipe, (s) Freewheel clutch, (t) Small break piston, (u) Relief valve, (v) Pilot valve, (w) (x) Oil pipes, (y) Electric contact, (z) Stop on table, (z_1) Gear on underside of table.

connection between lines " e " and " a_1," and shuts off the latter valve. At the same time, the slide depresses contact " f," which initiates the turning motion of the table ; but this latter effect can only take place when all six slides have arrived in the end position.

Control of the turn-table.

The turning movement of the table is under the control of an ordinary gear type pump acting in conjunction with a magnetically operated valve, as shown in Fig. 7. There is also a hydraulic power unit " o," the pinion of which engages with the large gear of the table. When the slides have reached their end position and are engaged in contact with " f," an electric circuit is closed, energising the electric motor " a " via a relay, thereby starting the pump " b." The valve " c " is now at the extreme right, so that the pressure oil delivered by the pump passes through line " d " to the right hand side of piston " e " (in contrast to the position shown in Fig. 7). The piston therefore moves to the left and moves pin " j " to the right via rack " f," push-rod " g " and lever " h." When the end position is reached, the same pressure oil passes through line " k " to act upon the r.h. side of piston " l," thus moving it to the left and pulling stop " m " to the left until it is disengaged from the corresponding stop " z " on the table. The oil is then passed through line " n " to the power unit which turns the table in the direction indicated in Fig. 7.

Shortly before the next table position is reached, another stop operates contact " p," and energises solenoid " q," which in turn retracts valve " c " to the left. Pressure oil now flows through line " r " to the left of piston " l," pushing it to the right and barring the way of table stop " z," while oil continues to flow through line " n " to the power unit. The locking pin " m " contains a small oil buffer " t," which, under the thrust exerted by the stop " z " discharges oil through the relief valve " u," to produce a braking effect large enough to absorb the kinetic energy of the table. At the same time, the oil pressure rises in the lines " r " and " n", overcomes the spring thrust of valve " v " and uncovers the inlet of line " w." Pressure oil is thus admitted to act upon the l.h. side of piston " e," moving it to the right and pressing the wedge " z " against the stop on the table. As the oil pressure is still on the increase, pressure is also transmitted to line " x " connecting with the six tool slides. This pressure impulse moves the pilot valve " b " (Fig. 6) into the accelerated speed position, thus setting the slides in motion. As the pressure increases further, contact " y " is closed, this in turn actuates the control for shutting-down the pump " b " and at the same time brings the valve " c " back to its home position at the extreme right by means of an auxiliary relay. As soon as the home position is reached, an auxiliary contact de-energises the auxiliary relay, and the mechanism is ready to effect the next turning movement of the table.

OPERATING ECONOMY OF PNEUMATIC AND STEAM FORGING HAMMERS.

By F. KNORR. (From *Stahl und Eisen*, Vol. 63, No. 22, June 3rd, 1943, pp. 433-437).

FOR reliability and ease of control of forging operation steam hammers have remained unexcelled for many years. These qualities have been fully retained in pneumatic forging hammers, which represent a subsequent stage of development. In view of the widespread employment of both types, a comparative study of their operating characteristics is of interest. This study is, however, limited to sizes up to a tup weight of 3000 kg., since this is the largest size to which direct electrical drive is applicable at present.

In judging hammer performance, knowledge of the entire range of blow strength from zero to maximum is important. Special consideration is therefore given to this point. The investigation extended to two new 500 kg. hammers in order to eliminate any factor of wear. At a later stage hammers which had been in service for some considerable time were also investigated, taking care to establish the influence of progressive wear upon hammer efficiency. Finally, comparative power consumption in actual shop practice was ascertained by forging of test pieces.

A typical pneumatic forging hammer equipped with controlling and hammer cylinders is shown in Figs. 1 and 2. This simple standard type has proven most satisfactory, but difficulties are sometimes experienced with the forging of large and unwieldy pieces. For this purpose, a heavy portal type, on the lines of steam hammer practice, has recently been developed.

Except in very special cases, pneumatic forging hammers are usually built for unit drive by means of an electric motor. The power transmission may be by belt drive from the motor pulley to the fly-wheel of the hammer, or by a reduction gear, or by a combination gear. While the belt drive is of particular simplicity, it has the disadvantage of requiring an expensive low-speed motor. Reduction gear drives are somewhat more complicated, but permit of the employment of a less expensive high-speed motor. As the additional expense incurred with the use of a slow-speed motor is relatively unimportant with hammers of small and medium size, the direct drive is frequently resorted

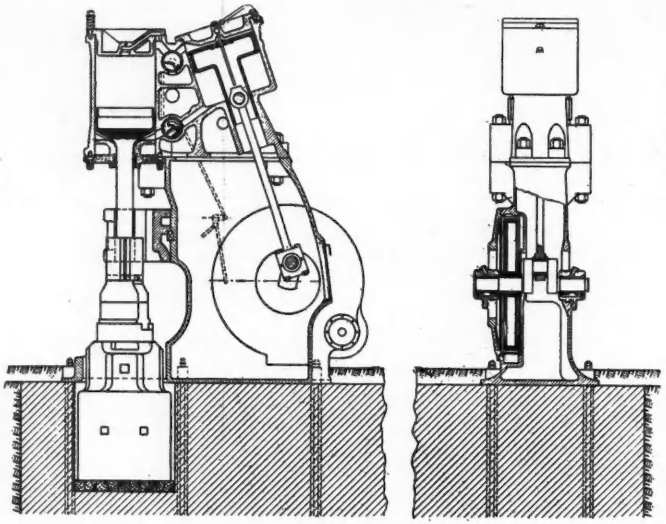


Fig. 1. **Sectional view of compressed air hammer with electric drive. (Eumuco type).**

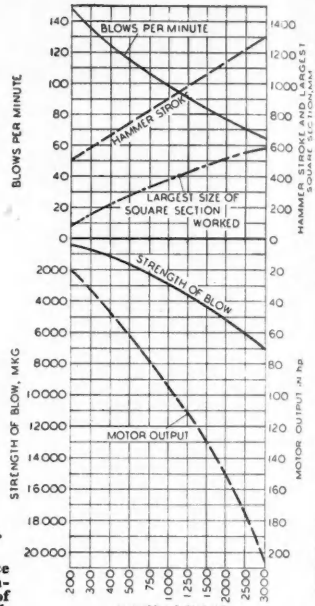


Fig. 3. **Performance characteristics of compressed air hammers of standard design with electric drive.**

to in these size ranges, while for large hammers a reduction gear is preferred.

The performance characteristics obtained with a series of normal pneumatic hammer sizes are shown in Fig. 3, hammer capacity being based on the forging of a material of 40-50 kg. per sq. mm. ultimate tensile strength (at room temperature). Excepting motor speed variations with varying hammer load, all these hammer sizes operate with an equal number of hammer blows per minute, irrespective of the length of stroke employed in the forging operation. Variation of hammer speed can, however, be easily obtained with the use of variable speed motors. For reasons of design, speed variations in excess of 30 per cent are not recommended; while a speed range extending from 15 per cent below to 10 per cent above normal hammer speed can be easily realised.

In the pneumatic type, control of the strength of the hammer blow is effected by means of two piston valves, shown in Fig. 4, which control the air volume supplied to the hammer cylinder. The time *vs.* stroke diagram reproduced in Fig. 5 shows the position of the pistons of both hammer and control cylinder through the working cycle. In this chart both the computed and the actual time *vs.* stroke characteristics (curves II and III respectively) of the hammer piston are given. During the upward stroke, the piston ring friction and gland friction reduce the hammer piston speed below the computed values. Also, the air compression process leads to an increase in the temperature of the air; this causes a certain amount of transmission of heat to the cylinder wall, hereby reducing the actual compression below the computed value. Thus both upward and downward-stroke hammer-speed, as well as the length of stroke, are below the computed values; and this of course affects also the effective force of blow. As the heat loss through the cylinder wall will rise with increased air temperature, that is, with the compression ratio, the hammer efficiency will necessarily increase with diminished max. air pressure during the working cycle.

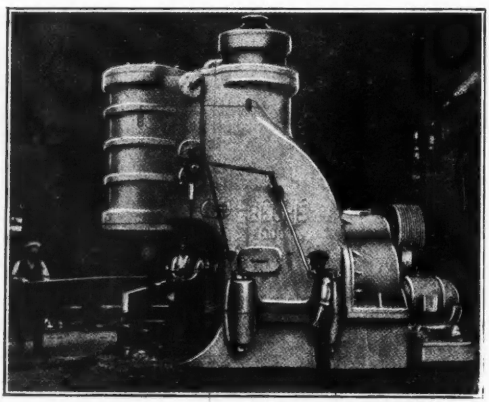


Fig. 2. **Compressed air hammer with 2,000 kg. tup. (Bêché type).**

Apart from this, particularly when considering continuous operation, care must be taken to prevent the air temperature from exceeding the maximum permissible operating temperature of the lubricating oil. The air temperature characteristic of a 500 kg. compressed air hammer is shown in Fig. 6. With a normal compressed air forging hammer of modern design, artificial cooling should not be necessary. This applies also to hammers employed for drop operations.

Water cooling cannot, however, be dispensed with in the case of high-speed hammers for continuous operation. This requirement has been found to arise from the fact that the heat transfer from the cylinder to the ambient air is too slow to prevent the failure of lubrication and consequent ultimate piston seizure. This danger recedes with intermittent operation; but in doubtful cases, artificial cooling will be found preferable.

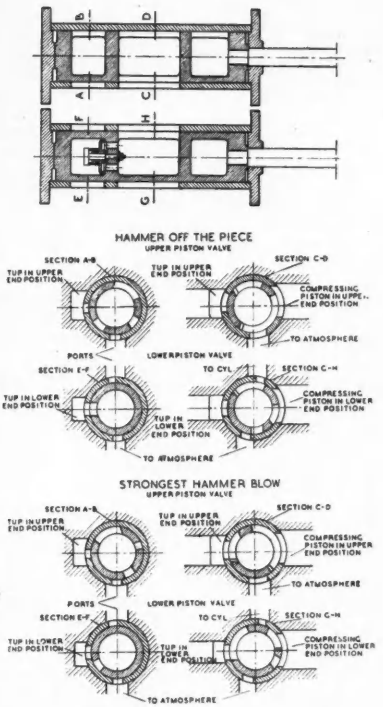


Fig. 4. **Operating scheme of piston valves.**

That the air hammer is not comparable to a compressor becomes at once obvious when it is realised that a compressor has a continuous throughput of fresh air; while in the hammer operating at its maximum strength of blow, the same air is used in the successive working cycles. Only at the end of stroke is sufficient fresh air admitted to make up for unavoidable leakage losses. Thus, while the fresh air admitted to the cylinder of a compressor absorbs a certain amount of heat from the cylinder walls, the air contained in the cylinders of the hammer will rise to the temperature of the cylinder walls, and the succession of working

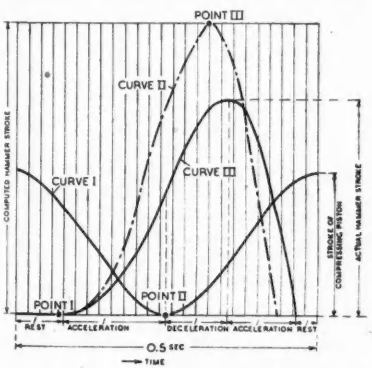


Fig. 5. **Travel *versus* Time characteristics of hammer piston and compressing piston.**

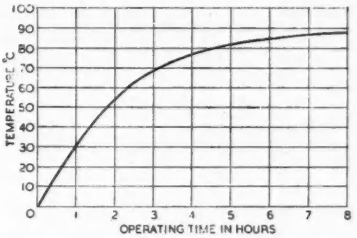


Fig. 6. **Air temperature *versus* operating time.**

cycles increases both the temperature of the air and of the cylinder walls.

Elevated air temperatures also effect a marked evaporation of lubricant; and this vapour escapes when the hammer is idling. In order to prevent adverse effects of the vapour upon the health of operating personnel, an air vent to atmosphere should be provided. It is natural that the thermal effects in the cylinders are of adverse influence upon hammer efficiency. With hammers of the series referred to in Fig. 3, efficiencies of 60-75 per cent obtain.

Indicator diagrams obtained with idling or with keeping the hammer off the piece, and with various strengths of blow, are shown in Fig. 7. Here it is seen that only the lower half of the compressing cylinder is effective when keeping the hammer off the work piece. When idling, the power consumption of the motor, comprised by the diagram area, is seen to be small compared to that registered by the diagrams taken at maximum strength of blow. Likewise, the power consumption recorded with light blows is small compared to that registered with the hammer exerting its full force. In drop forging, the rapid succession of hammer blows is of particular advantage, as this shortens the time of contact between work piece and die, and with it the time available for cooling-off. In addition to this, the heat imported by the hammer to the piece also increases with the frequency of the blows. The drop in temperature during the forging process therefore remains small. Service life of the hammer components will be satisfactory and fractures comparatively rare if the work pieces are properly heated and the hammer is not continually used for giving chamfering blows.

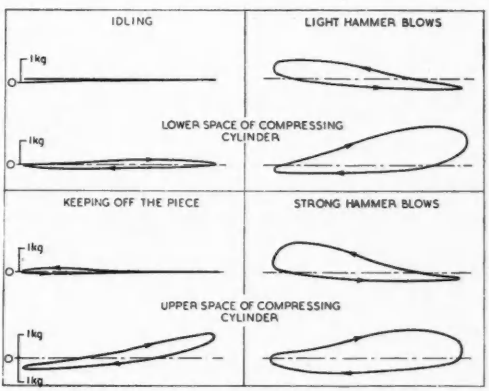


Fig. 7. **Indicator diagrams of compressing cylinder.**

THE AERODYNAMIC TURBINE VERSUS STEAM AND GAS TURBINES

By C. KELLER. (From *Escher Wyss Mitteilungen*, Vol. 15-16, 1942-43, pp. 20-41).

WHILE in steam power technique highly efficient operation with very large unit sizes has been achieved, internal combustion engines have heretofore suffered from enforced limitation in size. Similar considerations apply to the recently introduced combustion gas turbine of the constant pressure type. However, the operating principle of the aerodynamic turbine employing heated compressed air is now pointing the way in which the individual inherent thermodynamic and constructive advantages of the steam turbine and the internal combustion machine (engine or turbine) can be successfully combined.

The considerations upon which the development of the aerodynamic turbine is based are as follows :—

(1) The power unit shall be particularly suitable for cases in which large prime mover size is required to carry continuous load, as in stationary thermal power plants and in ships propulsion.

(2) Fuel requirements must not be limited to the use of liquid fuel, but the firing of gaseous fuels and, above all, coal must likewise be possible.

(3) Power generation must be of the nature of a continuous process in high speed turbines and not a cyclic process involving explosions and the operation of valves, igniting devices, etc.

(4) Thermal efficiency both at normal and partial load of the novel prime-mover must at least equal that achieved in modern steam plant or binary-plant practice.

(5) Superiority of the novel type plant must not be purchased at the expense of complicating plant design.

That steam is by no means the ideal medium from the thermodynamic point of view is amply proven by the fact that approximation to the Carnot cycle, or any equivalent cycle, is more difficult with steam as operating medium than it is with gas or air. Taking the upper cycle temperature as 600° C. and the lower as 30° C., theoretical Carnot efficiency is $\eta_c = 65.4\%$. If the steam and the aerodynamic cycle are based on initial pressures of 30 atmos., only 60.2% of the theoretically possible heat utilisation can be achieved with the steam cycle, while the aerodynamic cycle permits a utilisation of 89·1% ; and in the case of a gas turbine employing the same admission/exhaust pressure ratio of the aerodynamic unit ($p_1/p_2 = 4$), the corresponding figure will be only 75.8%.

Considering that the useful work available per unit weight of working medium is considerably larger with steam than with gas, this would at first thought appear an important advantage with regard to comparative unit size. This implication can be disproven at once by establishing unit size requirements on the basis of the required volume of the working medium. In the case of the steam turbine, steam volume is uncomfortably small in the initial stage, and extremely large at the exhaust. This is, of course, due to very large inlet/exhaust pressure ratio encountered in steam turbine practice, which, in the case of 30 atmos. initial pressure leads to a 200 fold volume increase at the turbine exhaust (assuming of course condensing operation). In contrast to this, the pressure ratios, and with them the adiabatic heat drops of the gas cycles, are greatly reduced, and turbines of these types therefore require fewer stages than a corresponding steam turbine. Also, the gas cycles admit of the use of higher initial volumes at the first stage, thus eliminating the employment of the very short blading required in the initial stages of the steam turbine. On the other hand, the gas turbine does not require the large blade length called for in the last stages of the steam turbine.

The aerodynamic turbine is further characterised by the fact that its thermal efficiency does not depend upon the absolute pressures employed. Unlike the case of the steam turbine, attainment of an increased thermal economy is not conditional upon the use of increased pressure, so that the pressure level can be chosen from the aspects of design alone. For the very largest unit capacities a pressure range between 20-50 atmos. has been found particularly suitable. Units designed for this pressure range are of remarkably small dimensions, without, however, being handicapped by the complexities of high steam pressure operation.

In actual practice, the principle of isothermal compression and expansion which forms the basis of the novel process can only be approximated. Although in principle an increase in the number of compression or expansion stages leads to a rise of efficiency, its rate of increase, while rapid at first, diminishes progressively with augmented stage number. At the same time, an increase in the number of stages is accompanied by unavoidable losses, apart from the complexity of piping connections engendered, so that in the balance a large number of stages does not result in any further rise in efficiency. It follows that with as little as one or two intercooling stages on the compressor side, and with direct expansion (or with possibly one reheating stage on the expansion side), the economic optimum is already attained. This also leads to simplicity of design.

The actual operating cycle will thus be as indicated in Fig. 1. Comparing this cycle with that of the gas turbine exhausting to atmosphere (Fig. 2), it can be seen that both cycles employ identical component parts. Thus, both cycles comprise 1 turbine, 1 compressor, and 1 heat-exchanger each ; while in the aerodynamic circuit, a gas heater takes the place of the combustion chamber employed in the gas turbine cycle. The only additional components of the aerodynamic cycle, as compared to the gas turbine cycle, are the pre-cooler and the intercoolers of the compressor. However, these coolers must likewise be incorporated in the gas turbine cycle if its efficiency is to be comparable to that of the aerodynamic cycle.

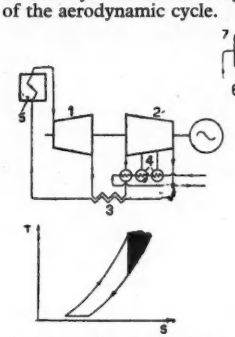


Fig. 1. **Aerodynamic turbine operating on the closed cycle.**

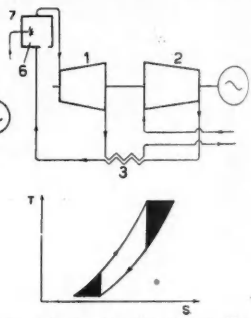


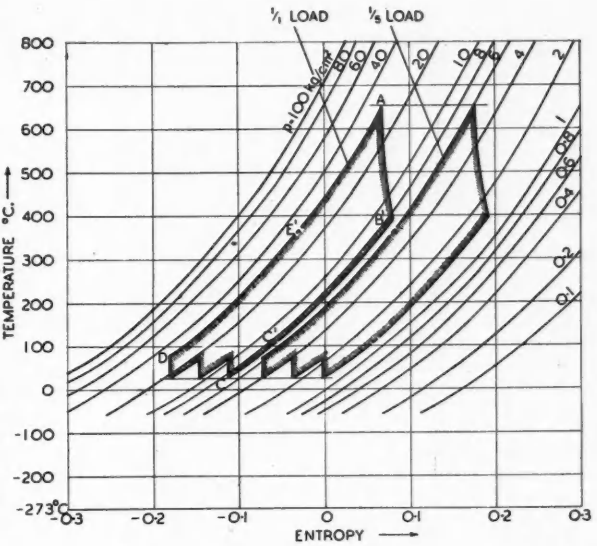
Fig. 2. **Combustion turbine operating on the open cycle.**

(1) Turbine, (2) Compressor, (3) Heat exchanger, (4) Inter-cooler, (5) Air heater, (6) Combustion chamber, (7) Fuel supply.

The cycle conditions to be chosen in present-day practice are charted in the temperature-entropy diagram reproduced in Fig. 3, which also indicates the pressure and temperature conditions that will prevail in the component parts of the circuit. This diagram also gives the cycle conditions established at partial load with the change in density of the working medium. It is shown that with partial load, the absolute pressures decrease proportionally with the load, with both the temperatures and the internal efficiency of the cycle remaining unchanged. Since at fractional load, the pressure stresses are correspondingly decreased from

Fig. 3. **Entropy-temperature diagram for air showing the aerodynamic cycle.**

AB^1 = expansion in turbine.
B^1C^1 = cooling in heat exchanger.
C^1C = pre-cooler.
CD = compression with inter-cooling.
DE^1 = heating of air in the heat exchanger.
E^1A = air heater.



their full load level, a slight increase in operating temperatures may be permissible. By making use of this possibility, the relative increase in mechanical losses of the plant can be compensated for, so that total plant efficiency will fall only slightly over a wide range of load. On the other hand, overload capacity of the plant can be secured by admitting compressed air into the cycle; and lest this should lead to excessive stress at normal operating temperature, the pressure increase at overload may be compensated for an appropriate decrease in operating temperature. This facility of output variation by a change of the circuit pressure, makes it particularly easy to cover a wide range of plant capacity with a very small number of unit sizes. Thus a capacity range of 2,000 to 50,000 kW can be covered by eight different designs. Still fewer different sizes of auxiliary equipment will be needed as the required capacity of heat exchangers, coolers, etc., can easily be obtained by resorting to the parallel arrangement of standard sizes.

Owing to the density of the working medium, the dimensions of the turbines and compressors of the aerodynamic circuit are surprisingly small; and as the heat drops are only about ¼–1/5th of those prevailing in steam turbines, the number of turbine stages is small. The approximate size of units ranging from 12,000 to 100,000 kW capacity are shown in Fig. 4. According to the turbine and compressor type chosen, there will, of course, be slight variations in size. The fact that no control gear at all is required, makes aerodynamic turbine design a particularly easy task. Also, since a turbine is always laid out for full admission, casing design is greatly simplified.

Even where only moderate pressures of up to 30 atmos. are employed, single-shaft units can be employed for the very largest unit sizes. For large unit outputs compressors of the axial type, shown in

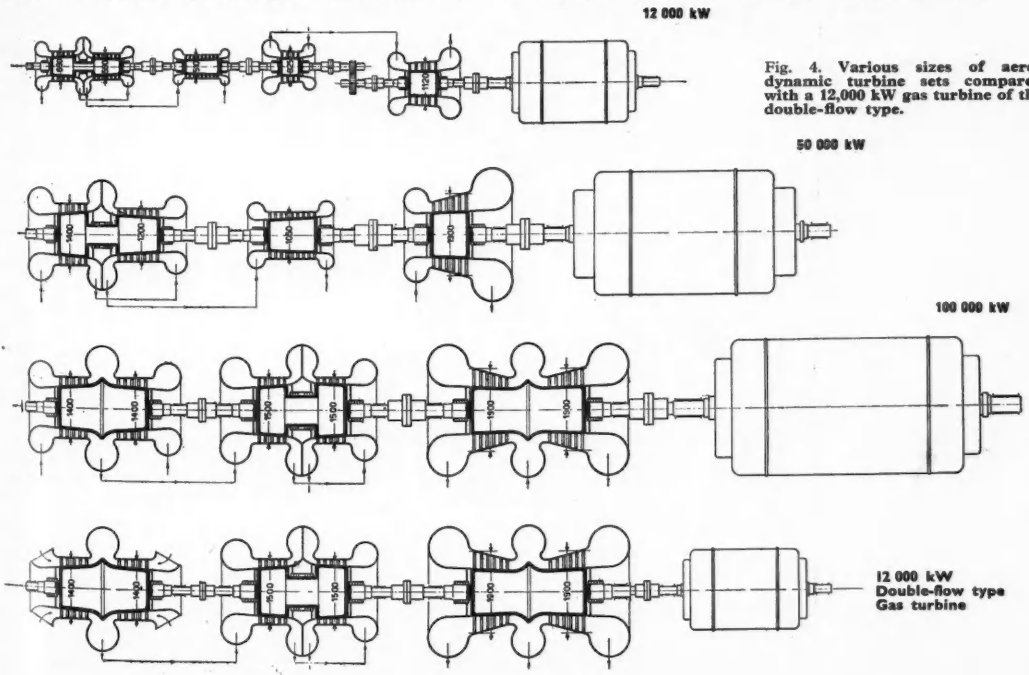


Fig. 4. **Various sizes of aerodynamic turbine sets compared with a 12,000 kW gas turbine of the double-flow type.**

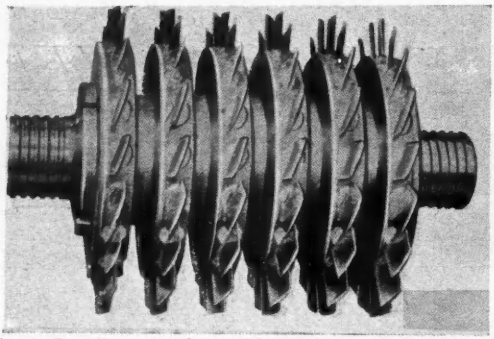


Fig. 5. **Impeller part of a multi-stage axial compressor.**

Fig. 5, are particularly suitable. Because of the smallness of the heat drop employed, especial attention has to be paid to achieving minimum inlet and exhaust losses. Since all turbine control organs are absent (Fig. 6), this requirement can be met by making the turbine, as it were, part and parcel of the gas conduit proper, as indicated in Fig. 7. From the latter engraving, it will also be seen that the small turbine dimensions permit the use of double walled casings with the inter-space filled with thermal insulating material. In this design, the actual pressure part is formed by the cold outer casing, with the thin inner casing holding the insulation in place and guiding the flow of the working medium. The entire stationary blade system is suspended within the casing and held in place in such a manner that the outer casing itself at the most is subjected to the exit temperature of the working medium, no matter how high the initial temperature of the latter may be. This does away with the need for a turbine casing of high heat resisting material.

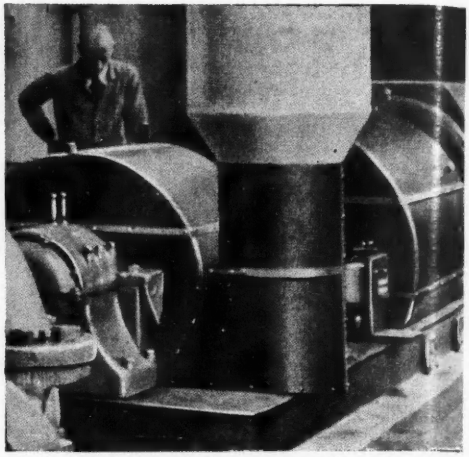


Fig. 6. **Admission end of an aerodynamic turbine showing absence of control gear and valves.**

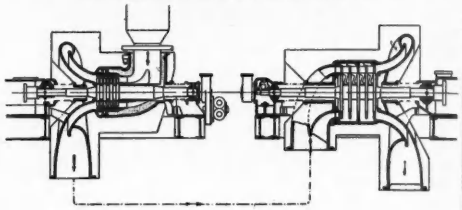
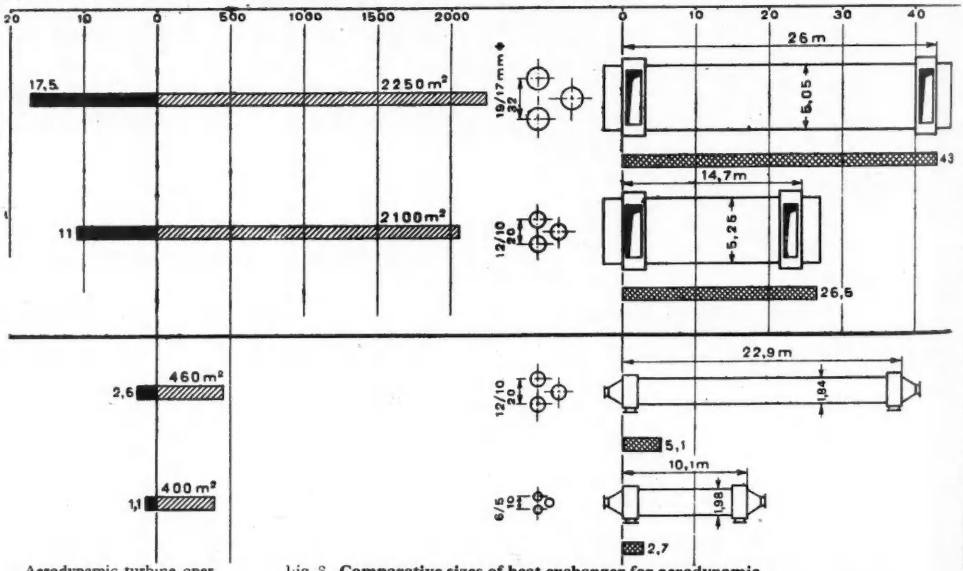


Fig. 7. **Stream-lined design of 2-casing aerodynamic turbine unit with insulated first casing.**

THE HEAT EXCHANGER.

Convection heat transfer from flowing air and gaseous media is known to be poor under normal conditions. But this situation is radically changed if elevated working pressures are employed. This is strikingly illustrated by Fig. 8, which shows the heat



Aerodynamic turbine oper. pressure 28.7 atmos.

Fig. 8. **Comparative sizes of heat exchanger for aerodynamic and open-cycle combustion gas turbines.**

exchanger surface requirements and dimensions of the closed aerodynamic system as compared to the open gas turbine cycle. This diagram is based on a plant size of 12,000 kW with a working pressure range of 7–28 atmos. for the closed, and of 1–4 atmos. for the open cycle. The heat exchangers are assumed to be of the tubular type working in counterflow fashion. The fact that the aerodynamic cycle is seen to require less than one-fourth of the heat exchanger surface called for by the open gas turbine cycle, is due to both the higher density of the working medium and to the increase in specific heat transference by convection, with increased density of the working medium. To complete the comparison of relative heat exchanger surface requirements and of heat exchanger dimensions for the two cycles, Fig. 8 includes analogous data based on the use of heat exchanger tubes of identical diameter for the two types.

As, unlike the gas turbine cycle, no flue gas enters the aerodynamic cycle itself, no soot deposits can occur on the next heat exchanger surface. Tube diameter can therefore be chosen as small as is considered desirable The specific heat transfer coefficient is hereby further increased.

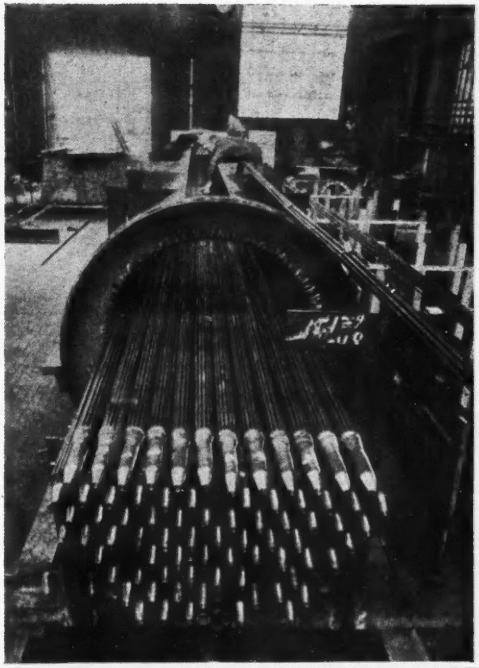
While in principle any kind of surface arrangement can be used in the heat exchanging device, both pressure conditions and considerations of investment cost make it preferable to use a tubular surface. A counter-flow heat exchanger of this type, in course of assembly, is shown in Fig. 9. In this design the high pressure air is passed through the tubing, so that the exchanger shell is exposed only to low pressure. It is frequently found desirable to combine the water cooled pre-cooler with the heat exchanger, particularly as the former is located on the downstream side of the latter. This can be accomplished with use of the design shown in Fig. 10, thus eliminating pipe connections and attending pressure loss.

THE AIR PREHEATER.

In the aerodynamic cycle, the air heater is the equivalent to the boiler of the steam cycle. Thus, the heat of the fuel is imparted to the working medium by means of the air heater surface, hereby precluding contact of the flue gases with the prime-mover plant proper. In contrast with the boiler plant, no evaporation takes place in the air heater, its role being limited to merely adding a requisite amount of heat to the working medium admitted to the air heater at 300–400° C. As this constitutes a straight heating process, no vapour spaces in the shape of drums are required. Since the air to be heated is under pressure, the air heater will preferably be of the tubular type, with the heating surfaces sub-divided into radiant and convection parts. As no water is passed through the apparatus, there is no objection to out-door installation, as there is no danger of freezing.

Due to the aforementioned fact that with the air under pressure favourable convection heat transfer coefficients obtain, tube metal temperatures can be kept within the range permissible with the commercially available alloyed materials. Theoretically, it would, of course, be possible to increase the heat transfer from tube wall to air to such a point that the tube wall temperature approximates that of the air, but this cannot be realised in practice because of pressure drop considerations. Actual air velocities,will, therefore, be so chosen that the tube metal temperature remains within the permissible range. Based on this consideration, the use of graduated tube diameters has been adopted, the tube diameters being stepped up as the air temperature increases, as shown in Fig. 11.

Thorough investigation of these problems has resulted in the development of a number of basic arrangements, which, with a minimum of pressure drop, enable the attainment of air temperatures far in excess, not only of present-day steam temperatures, but also of anything that has so far even been contemplated as a future possibility in the use of superheated steam. As the flue gases leave the air heater at relatively high temperature, their waste heat can be used to



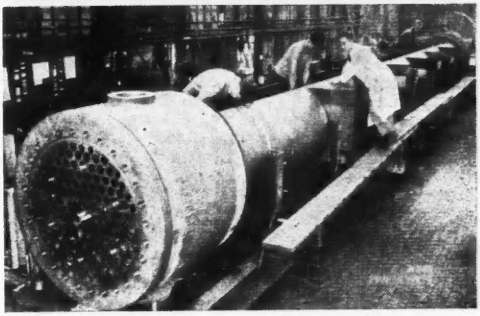


Fig. 9. **Heat exchanger in course of construction.**

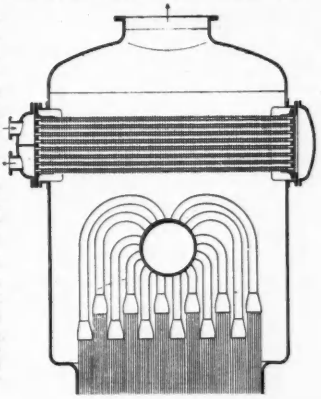
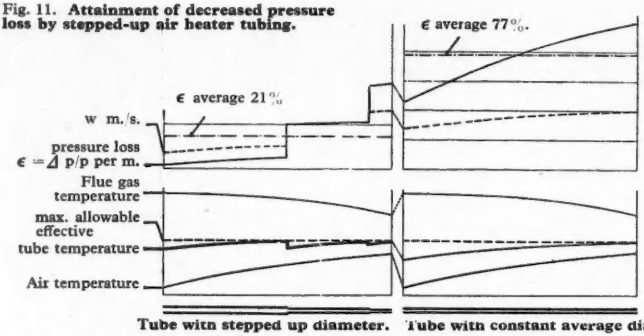


Fig. 10. **Combined heat exchanger and cleanable pre-cooler (patented).**

Fig. 11. **Attainment of decreased pressure loss by stepped-up air heater tubing.**



advantage for preheating of the combustion air. This is particularly desirable in pulverised fuel burning plants where the use of preheated secondary air will serve to reduce the time required for completing combustion. Any increase in combustion temperature must necessarily increase the thermal loading of the tubing ; but in this respect the employment of flue gas recirculation provides a simple means for controlling furnace temperature at practically any required level.

Mention has already been made of the fact that further decreases in metal temperature can be obtained with the use of suitable flow arrangements (parallel current, counter-currents, or cross-current) in the individual parts of the heater. One of the many arrangements that are available for this purpose is shown in Fig. 12, where the counter-current of the heated air is divided into two parallel circuits, the one exposed to the hottest gases being arranged in parallel flow, while the other which is swept by the cooler gas, is dispositioned in counter flow fashion. Besides limiting the metal temperatures in parts of greatest head load, double stream arrangement also secures a decrease in the total pressure drop of the system. Referring to the surface versus temperature diagram included in Fig. 12, it will be seen that only small portions of the air heater need be built of high heat resistant materials. Attention is also drawn to the fact that because of the small circuit pressure employed, tube stresses will be no more than 1–3 kg./sq. mm. even with very thin tubing. Stresses of this order are considerably below the actual creep strength of modern high temperature steels.

CONTROL OF THE AERODYNAMIC POWER PLANT.

In contrast with the steam or gas turbine cycle, the closed aerodynamic power cycle alone makes possible continuous loss-free load control at constant speed over the load range, since its output control is effected by varying the density of the working medium, that is, the circuit pressure. A scheme for automatic control based on the variable density principle is outlined in Fig. 13.

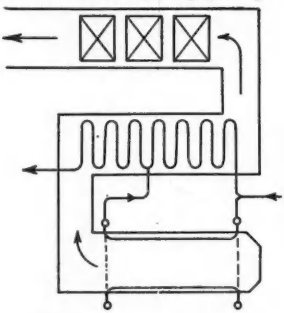


Fig. 12. **Patented air heater design in which the use of special high temperature steel is limited to a small portion of the heating surface.**

Here the control gear is actuated by the speed governor (5) which enables admission of the working medium from the high pressure storage (8) into the working circuit, or *vice versa*, discharge of working medium from the circuit into the low-pressure storage (9). A by-pass valve (10) is also provided, which permits direct flow from the high pressure to the low pressure side of the circuit.

The control process may be exemplified as follows: An increase load will be attended by a decrease in the speed of the turbine, causing the speed governor (5) to open the inlet side V_H of control valve (7). Working medium will therefore flow from the high pressure storage into the circuit, hereby increasing the power output of the latter. The required correlation of the speed actuated governor impulse with the load is obtained by the coordinating organ (6), which transmits the governor impulse to the control mechanisms of valve (10). *Vice versa*, decreasing loads lead to the discharge of working medium via valve (7) to the low-pressure storage V_N, hereby reducing the power output by the required extent.

The largest portion of the working medium in the closed cycle is contained in the heat exchange apparatus, that is, in the heat exchanger, air heater, and pre-cooler, partly under the pressure p_H of the high pressure section and partly under the pressure p_N of the low pressure section. Introduction of appropriate amounts of additional working medium from the storage container into the working circuit when the load increases, is effected on the high pressure part of the system so that the available power is augmented. In the actual control process, this supply of high pressure working medium to the circuit will lead to a momentary increase in upper working pressure, hereby increasing the pressure ratio p_H/p_N. With this, the quantity of working medium passing through the turbine is increased, while the compressor delivery is decreased, hereby making useful power available. *Vice versa*, a decrease in plant load leads to a release of working medium on the high pressure side, which results in an instantaneous drop in pressure p_H and a decrease in the ratio p_H/p_N, as p_N remains momentarily unchanged. With the resultant decrease in the flow rate of power medium through the turbine, its output is correspondingly reduced, while the delivery of the compressor is increased. The useful power output is, therefore, at once considerably decreased. It is computed that with rapid discharge of the working medium, the power output of the aerodynamic plant can be reduced from full load to zero by discharging slightly less than 20% of the working medium contents of the circuit.

(*To be continued in next issue*).

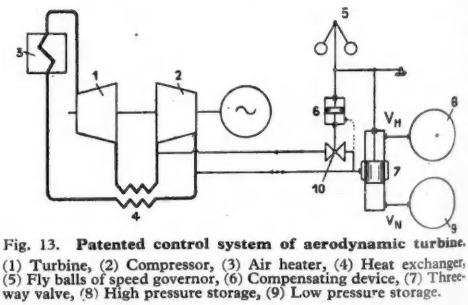


Fig. 13. **Patented control system of aerodynamic turbine.** (1) Turbine, (2) Compressor, (3) Air heater, (4) Heat exchanger, (5) Fly balls of speed governor, (6) Compensating device, (7) Three-way valve, (8) High pressure storage, (9) Low pressure storage.

THE SOVIET W 2 TANK DIESEL.

By AUGUSTIN. (From *MTZ Motortechnische Zeitschrift*, Vol. 5, No. 4/5, May, 1943, pp. 130-139).

THE various types of engines used in the smaller Soviet tanks are well known and present but little technical interest, since these are based on the old four-cylinder Ford AA or BB engines, the Ford V-8 engine, or the air-cooled Armstrong-Siddeley horizontal four-cylinder model. It is surmised that the Ford AA and the Armstrong-Siddeley engine were taken over at a date when these designs had already been abandoned by the original makers. But this does not, of course, mean that these designs are inferior. On the contrary, the Ford engine in particular, in spite of its antiquated design, still represents an absolutely reliable and, above all, simple type, thus meeting the two most important requirements of an engine for the armed forces.

Quite different is the position with regard to the powerful engine developed in the Soviet Union for use in heavy tanks and partly also for tractors. The requirements of a tank engine, i.e., small space and low weight, led to a far-reaching adoption of aircraft engine features. Actually, the first engine used in heavy Russian tanks was a slightly modified Otto-cycle aircraft engine.

However, when the introduction of a diesel engine for tank propulsion was decided upon, the Russian designers were faced with the task of developing an engine of this type, since no foreign prototype was available. In this task, the Soviet engineers proceeded along the path chosen on several previous occasions, of utilising several existing prototypes by either taking them over completely or following the models as closely as possible, incorporating whatever alterations were deemed advisable. In the case considered, engine design was based on the Hispano-Suiza twelve-cylinder aircraft engine. As the latter operates on the Otto-cycle, its conversion into a diesel design was required. This was effected by adapting the diesel design of the French firm "Coatalen," which design was first adapted by the Krupp-Germania yard for their marine diesels. As fuel injection pump, the Bosch design was chosen. With the exception of its external features, the Russian model is an exact replica of the Bosch pump; its working parts being actually interchangeable with those of the German make. Such an adoption of foreign designs obviously has the advantage that the best design available can be chosen. In addition to saving time and expense, and to short-circuiting "teething" troubles, this procedure makes it possible to concentrate every effort upon the production problem proper.

Data on the salient engine components are given in Table I., while the engine characteristics are charted in Figs. 1 and 2. Fig. 1 gives engine output, torque, and fuel consumption at rated engine capacity, while Fig. 2 shows the corresponding characteristics based upon an upper engine speed limit of n = 1800 r.p.m., engine

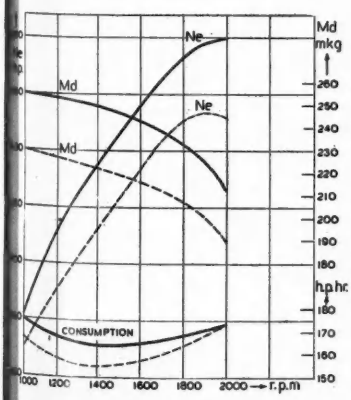


Fig. 1 Engine Characteristics.

Full lines: Adjusted to maximum output. Combustion not smoke-free. Injection pump adjusted to maximum delivery.

Dotted lines: Adjusted to smoke-free combustion.

Fuel: Diesel oil, spec. gravity 0.831. Barometer: 751mm. Cooling water: 80 deg. C. Air: 28 deg. C. Lubricant: 78 deg. C.

TABLE I.

DATA ON SOVIET TANK ENGINE W2.

Cylinder arrangement : V-60 deg.
Operating cycle : Four-stroke diesel
Cooling medium : water
Number of cylinders : 12
Bore : 150 mm.
Stroke : 186.7 mm. (180 mm.)
Swept cylinder volume : 3,297.6 cu.cm. (3,179.25 cu.cm.) per cylinder = 38,861.0 cu.cm. total volume
Combustion chamber volume : 236 cu.cm.
Compression ratio : 1:15 (1:15.8)
Fuel consumption : 155 g per h.p hour (with conditions of smokeless combustion and normal average load)

Output.
Specific output per litre of cylinder volume swept at max. output and 2,000 r.p.m. : 15.42 h.p. per litre
Normal cont. capacity at 1,900 r.p.m. with m.e.p. = 6.1 kg per sq. cm. : 500 h.p.
Max. capacity at 2,000 r.p.m. with m.e.p. = 6.9 kg per sq. cm. : 600 h.p.
Max. smoke-free capacity at 2,000 r.p.m. with m.e.p. = 6.18 kg per sq. cm. : 535 h.p.
Fuel consumption at max. smoke-free capacity : 167 g per h.p. hour.

Cylinder.
Material : Light metal
Liner : Wet type, steel, surface hardened

Piston.
Material : Light metal
Weight incl. rings and gudgeon pin : 3.845 kg.
Length : 120 mm.
Number of rings : 6
Height of individual ring : 2.5 mm.
Wiper ring height : 5.0 mm.
Gudgeon pin : Floating, steel, 48 mm. dia.
Mean piston velocity at 2,000 r.p.m. : 12.25 mm. per second

Connecting rod.
Length between bearing centres :
Main rod : 320 mm.
Auxiliary rod : 240 mm.

Crankshaft : Steel, not hardened, 8 main bearings
Crank webs : 161 mm. dia., 24 mm. width

Camshafts : Four, arranged on cylinder heads

Crank case : Light metal, weight 150 kg. incl. all stud bolts

Flywheel : 872 mm. dia., 182 mm. width, 180 kg. weight

Fuel pump : Type NKI
Injection pressure : 200 atm. g.

Electric heating plugs : none

Starter : 24 V, 15 h.p., weight 45 kg.

Generator : 24 V, 1.0 KW, weight 36 kg.

Overall dimensions of engine : Greatest length 1,450 mm.
Greatest width 900 mm.
Greatest height 1,000 mm.

Total engine weight : 811.7 kg.

Unit engine weight : 1.35 kg. per h.p.

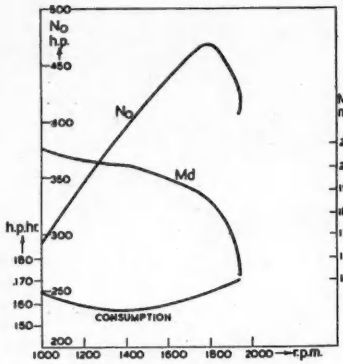


Fig. 2. **Engine characteristics.**
Speed limited to 1,800 r.p.m.
Limited delivery of injection pump.
Barometer: 758mm.
Cooling water: 80 deg. C.
Air: 21 deg. C.
Lubricating Oil: In—80 deg. C. Out—95 deg. C.

output in this case being 470 h.p. In connection with Fig. 2, it should be noted that engine power and engine speed are delimited in accordance with the type and weight of vehicle in which a particular engine is installed. This naturally has a favourable influence upon service life and reliability of the engine. Likewise, this measure serves to reduce the fuel consumption considerably and to enlarge the radius of action correspondingly.

From the very favourable torque characteristic, it is seen that in both cases maximum torque is obtained with less than 1,000 r.p.m. This means that a wide range of engine speed can be covered without necessitating gear-shift. Furthermore, the specific rate of fuel consumption must be considered extremely good and equal to that of the best German engine types. As shown by tests, all temperatures such as that of oil, water, etc., remained within normal limits.

In consideration of its intended purpose, the engine is designed with a view to obtaining lowest engine weight and smallest possible dimensions; but the requirements of sufficient service reliability have by no means been neglected. The exterior appearance of the engine is characterised by its remarkable neatness, and both cast and machined parts testify to the most careful workmanship. The fact that the piston alone bears no fewer than 22 inspection marks can be taken to indicate most rigorous inspection and acceptance procedure.

The disposition of the engine in the tank is well chosen, installation and removal of the engine being easily effected by means of a crane. The oil and water coolers are arranged alongside the engine, so that the cooling air passes first over the two radiators and then over the engine itself. Ventilation of the crew compartment is directly coupled to that of the engine compartment. Since both the chain track drive and the engine are arranged at the rear of the tank, the drive system is especially simple, the gear box being directly flanged to the engine.

A plan view of the crank case is given in Fig. 5. It consists of two parts bolted together at crankshaft centre-line. Its composition, given in Table II., corresponds to the German G Al-Si-Mg alloy. The lower part, made to form the oil-sump is made as low in weight as possible, while the upper part is of robust construction to sustain the forces acting upon it. Reinforcement by ribbed construction is provided. Precise assembly of the two parts is assured by means of locating pins. External surfaces are left unmachined, while the internal surface is covered with a brown varnish. The 7 journal bearings of the crankshaft are carried on an equal number of steel brackets bolted onto the upper part of the case. For fixing laterally the upper halves of the journal bearings and absorption of axial thrust,

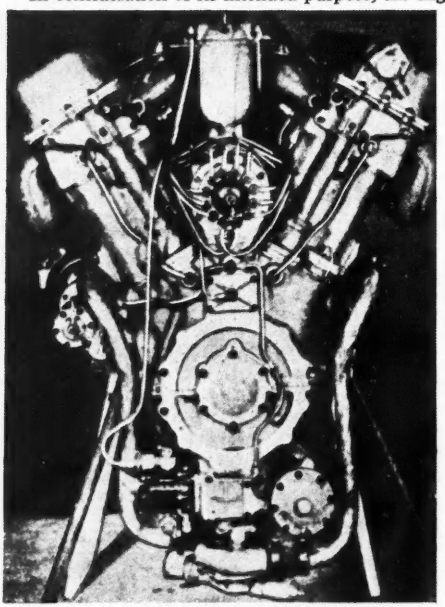


Fig. 3 **View of Engine.**
Top: Fuel filter and compressed air distributor. Left hand side: Generator: Left hand bottom: Fuel pump. Right hand side: Oil filter. Below it, cooling water pump, and behind that, lubricating oil pump.

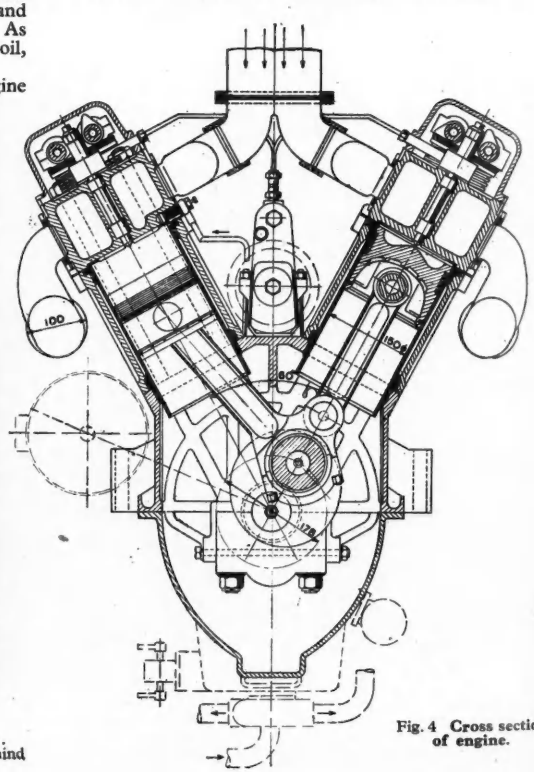


Fig. 4 **Cross section of engine.**

TABLE II. **LIST OF MATERIALS.**

	C %	Mn %	Si %	Cr %	Ni %	Mo %	Mechanical Properties	Heat Treatment
Main connecting rod .. (Aux. conn. rod similar)	0.15	—	—	1.49	4.28	0.35	116 k/sq. mm. UTS strength	Heat treated
Gudgeon pin	0.16	—	—	0.78	3.14	—	Core 124 kg/sq. mm. UTS strength Surface 63 Rockwell	Case hardened to 0.7 mm. depth
Cam shaft	0.1	—	—	0.4	2.16	—	Cam surface 61-63 Rockwell, core 67 kg/sq. mm. UTS	Case hardened to 0.7-1.0 mm. depth
Crankshaft	0.19	—	—	1.39	4.62	—	Scleroscope hardness 60-80 on journal surfaces	Flame hardened
Cylinder liner	0.35	—	—	1.46	—	0.45	Core 97 kg/sq. mm. UTS Surface 687 Vickers hardness	Surface nitrided to 0.44 mm. depth Water swept surface cadmium plated
Piston rings	2.82	1.21	1.57	0.16	0.47	—	Hardness 240 Brinell	Pearlitic
Exhaust valves	0.42	—	2.0	11.5	—	0.78	97 kg/sq. mm. UTS	Heat treated
Admission valves ..	0.18	—	—	1.07	3.77	—	92 kg/sq. mm. UTS	Heat treated

	Cu	Si	Fe	Mn	Mg	Al	Mechanical Properties	Remarks
Crankcase	—	9.78	0.28	0.37	0.32	bal.	27.4 kg/sq. mm. UTS 22.5 kg/sq. mm. 0.2 Y.P. 3.2% Elongation 83 Brinell hardness	casting
Cylinder block	1.1	4.88	0.38	0.13	0.47	bal.	24.3 kg/sq. mm. UTS 19.8 „ 0.2 Y.P. 1.2% Elongation 88 Brinell hardness	casting
Cylinder head	—	9.52	0.28	0.39	0.25	bal.	19.5 kg/sq. mm. UTS 16.0 „ 0.2 Y.P. 1.4% Elongation 79 Brinell hardness	casting

	Cu	Si	Fe	Mn	Mg	Al	Zn	Sn	Pb	Mechanical Properties	Remarks
Cylinderhead gasket ..	0.0	0.28	0.24	0	0	bal.	0	—	—	34 Brinell Hardness	Rolled Al sheet
Oilpump body	0.0	7.32	0.40	Traces	0.32	„	—	—	—	101 „ „	Cast alloy
Bush of small end bearing	87.75	—	—	—	—	—	1.7	10.0	0.51	92 „ „	„ „
Outlet-valve stem-guide	85.1	—	2.72	1.86	—	10.3	—	—	0	200 „ „	Rolled Bronze
Inlet-valve stem-guide	84.7	—	2.3	1.83	—	11.1	—	—	0	185 „ „	„ „
Piston ..	2.23	1.26	1.6	0.19	2.17	bal.	0	1.22 Ni	—	129 „ „	Pressing
Bush of big-end bearing, lead bronze ..	67.9	—	—	—	—	—	—	—	3.2	36 „ „	
Main bearing	71.4	—	—	—	—	—	0.48 Ni	—	28.4	28 „ „	

abutting blocks cast integral with the crank case are provided. At the front end, the casing is sealed by means of a sealing ring bush carrying three rings of 24 mm. height, the bush being carried on the shaft. The two camshaft drives (of the bevel gear type) and the generator drive are arranged at the other end. The two cylinder blocks are secured to the casing by means of 14 tie bolts as shown in Fig. 5. Four locating pins are provided to ensure proper fitting of the cylinder blocks to the top surfaces of the casing, the latter surfaces enclosing an angle of 120 deg. (corresponding to the 60 deg. V-arrangement of the cylinders). The front main journal bearing also incorporates a thrust bearing which takes the axial thrust acting on the crankshaft. The lower halves of the journal bearings are of the steel backed type, the 5 mm. thick steel shell containing a thin lead-bronze lining of only 2 mm. thickness. The two bearing halves are lined up by means of a locating pin of 9 mm. dia. Only the lower half of the extreme journal bearing is provided with a radial oil groove of 6 mm. width and 4 mm. depth for oil supply to the upper half of the bearing.

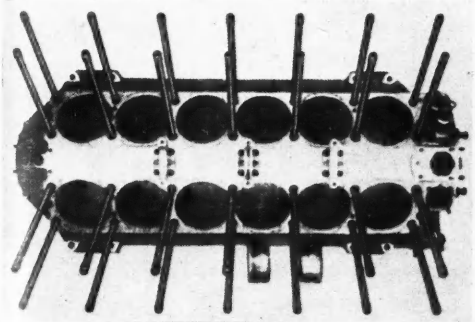


Fig. 5 **Crank case, Top view.**
(In the middle three bearings for the injection pump.)

The engine mounting on the frame is of the four-point support type, each support having two bolt holes of 16 mm. dia. The forged crankshaft of 1165 mm.

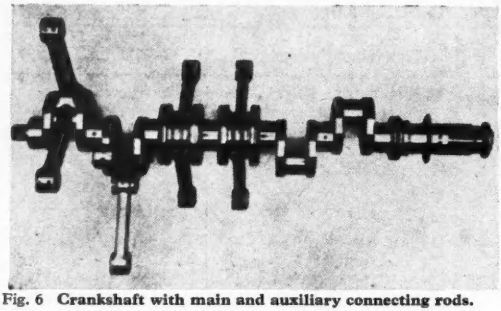


Fig. 6 **Crankshaft with main and auxiliary connecting rods.**

length (over the extreme bearings) is of the hollow type made with Chrome-Nickel steel, its weight being less than 70 kg. There are six cranks arranged in three planes at mutual angles of 120 deg. In order to obviate the necessity for balancing weights, circular crank webs are used, thus exactly following the Hispano-Suiza design. This secures the advantage of simplified manufacture and shortened machining time. The bearing surface of the crank pin is only slightly surface-hardened and can be cut with a file. The main bearing journals are generally drilled with one oil hole of 4.5 mm. dia. Only the extreme journal has three oil holes of 8 mm. dia. peripherically arranged at 120 deg. to each other; these holes serve for the admission of the lubricating oil into the hollow crankshaft and thus to the various points to be lubricated. All crank pins have 5 mm. dia. oil holes drilled at a leading angle of 55 deg. Oil supply of the individual bearings is effected by means of rolled-in straight oil pipes tapping the main oil stream. Oil supply from the main bearing to the crank pin takes place through two drilled oil holes of 8 mm. dia. In order to avoid oblique drill holes, the webs are drilled through and the extreme ends of the bore are closed by plugs. The seat of the gear flange is of the serrated type used for aircraft engines.

There are 6 forked connecting rods, with the main rods serving the l.h. and the auxiliary rods (of the oscillating type) serving the r.h. cylinder row. The design of the connecting rods closely resembles that of the French prototype, the connecting rods having I cross section, while the lower part of the bearing is reinforced with 4 ribs, being held in place by six polished stud bolts of 11 mm. dia. The latter are secured by drilling. Accurate positioning of the bearing halves is safeguarded by the provision of locating pins. The bearing halves consist of 2.5 mm. thick steel shells with 3 mm. thick lead bronze lining. There are no oil grooves, but the upper shell is drilled for oil supply of the big end bearing of the auxiliary connecting rod. The gudgeon-pin eye has a bronze bush secured in place by a hollow rivet. Both bush and eye are drilled with five

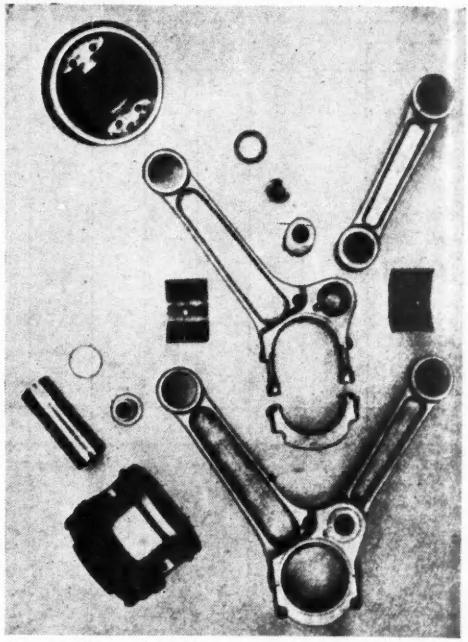


Fig. 7 **Main and auxiliary connecting rods, piston, etc.**

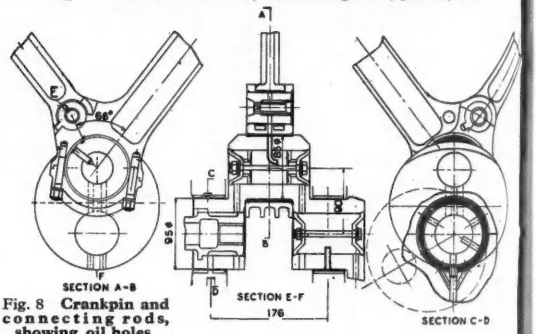


Fig. 8 **Crankpin and connecting rods, showing oil holes.**

oil holes in order to safeguard ample oil supply. The big end eyes of the auxiliary connecting rods appear somewhat small compared to the general dimensions of the rods, which are likewise of I cross section.

SEVERAL PROBLEMS ON TORSIONAL VIBRATION IN INTERNAL COMBUSTION ENGINES.

By Dr.-Ing. Kurt Haug, VDI, Berlin. (From *ATZ Automobiltechnische Zeitschrift*, Vol. 45, No. 15, 10th August, 1942, pp. 407-414.)

In order to prevent resonance, the frequency of the forces acting on the crankshaft must not coincide with the natural frequency of the crankshaft system. The excitation forces are given by the torsion force diagram in Fig. 1. The harmonic torsion forces are obtained by means of Fourier's analysis; their frequency is given by $\Omega_x = x\,\omega_0$, where ω_0 is the mean angular velocity of the crank pin at a particular speed, and x is the number of load cycles per revolution.

With two-stroke engines x can be, strictly speaking, 1, 2, 3, 4, etc., while with four-stroke engines it can be $\frac{1}{2}$, $1\frac{1}{2}$, 2, etc., as the total cycle takes place during two revolutions of the crankshaft.

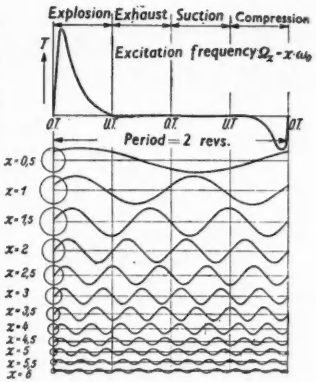
Thus, knowing the frequency of the excitation forces, it is necessary to determine the natural frequency of the crankshaft system, in order to find the conditions of resonance.

THE VIBRATING SYSTEM.

The system consists of the crankshaft, connecting rods, pistons, the flywheel with the clutch, and sometimes of a vibration damper mounted at the free end of the crankshaft.

A complicated system like this must be reduced to a simpler equivalent system to make possible a mathematical treatment of the problem (Fig. 2). One can

Fig. 1
Harmonic Analysis of Torque diagram.



obviously represent the system by a simple, elastic shaft without inertia, on which separate discs are mounted corresponding to the crank drives, flywheel, etc. If it is possible to replace the actual system by the system described above, then the determination of the torsional vibrations is relatively simple, as they can be expressed by linear differential equations with constant coefficients.

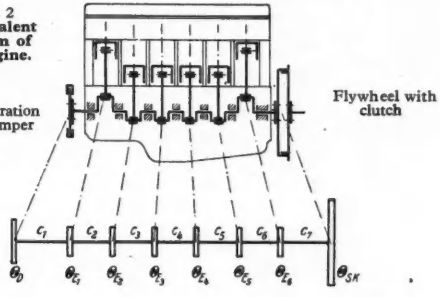
The torsion energy and the kinetic energy of both systems must be the same, so that the equivalent system has the same vibrational properties as the actual system. In the case of equal amplitudes, the kinetic energy of both systems is equal, if the moments of inertia of the rotating masses are the same. The torsion energy is the same if the torsional stiffnesses of the sections of the shaft between the rotating masses are equal.

In constructing the equivalent system, therefore, two questions have to be answered :—

1. What are the moments of inertia of the discs ?

2. What are the torsional stiffnesses of the various shaft sections ?

Fig. 2
Equivalent System of an engine.


Vibration damper
Flywheel with clutch
c_1 c_2 c_3 c_4 c_5 c_6 c_7
Θ_0 Θ_{c_1} Θ_{c_2} Θ_3 Θ_{c_4} Θ_{c_5} Θ_{c_6} Θ_{SK}

Θ Moments of inertia of discs of equivalent system.
c Torsional stiffness of crankshaft sections of equivalent system.

REDUCTION OF THE MASSES.

The moment of inertia of a disc must be such that its kinetic energy at any speed equals the kinetic energy of the corresponding crank drive, consisting of the connecting rod, piston, and gudgeon pin. Since the moment of inertia of the connecting rod Θ varies with φ, the arithmetical mean value should be calculated. In practice " Frahm's Approximation " is used instead of the exact arithmetical mean. The connecting rod is divided into the ratio of distances from the centre of gravity into an oscillating part ($m_{osc.}$) and a rotating part ($m_{rot.}$). It can be shown then that the kinetic energy (E) of the equivalent system is,

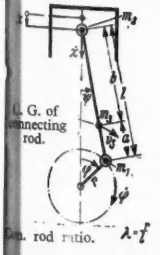


Fig. 3 **Crank drive. Determination of kinetic energy.**

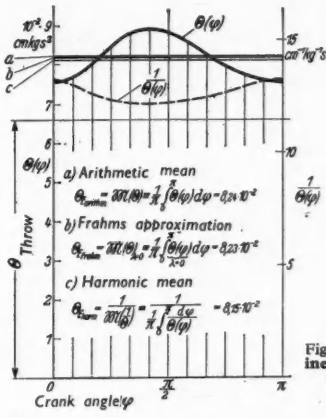


Fig. 4 **Moment of inertia of crank mechanism.**

$$\Theta E_{Frahm} = \Theta_{kr} + m_{rot.} r^2 + \tfrac{1}{2} (m_k + m_{osc}) r^2 \quad .. \quad (1)$$

where Θ_{kr} = M.I. of throw about the crank shaft axis, and m_k = mass of piston and gudgeon pin.

Mathematical examinations carried out by R. Grammel show that not the arithmetical mean but the harmonic mean

$$\Theta E_{harm} = \frac{1}{\frac{1}{\pi} \int_0^x \frac{d\varphi}{\Theta(\varphi)}}$$

should be used to obtain reasonably correct periods of vibration.

In Fig. 4 the curve Θ (φ) is given for a modern crank mechanism. The exact arithmetical mean, Frahm's approximation and the harmonic mean are shown.

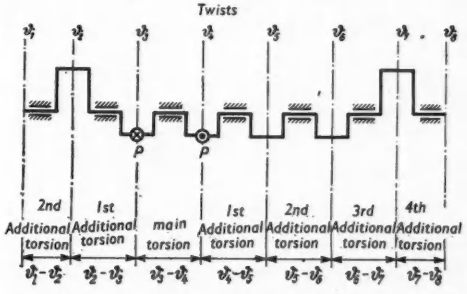
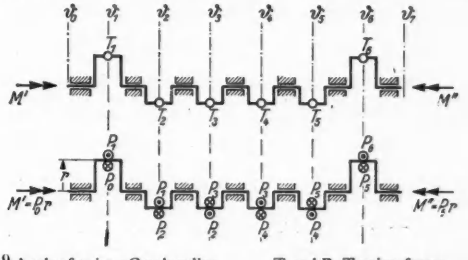


Fig. 5 **Main and additional torsions by torsion of 2nd kind.**



ϑ Angle of twist, r Crank radius. T and P Torsion forces. M Moments.

Fig. 6 **Resolving of load into pairs of forces.**

Comparison of the three values shows that the arithmetical mean and Frahm's approximation are only slightly higher than the harmonic mean—in the above example by only 1%. The periods of vibration, which are roughly proportional to $\sqrt{\frac{1}{\vartheta_g}}$, are larger by 0.5% only.

REDUCTION OF LENGTHS.

The question has to be answered: What is the important torsional stiffness in torsional vibrations of crankshafts? It has been customary, so far, to replace the crank throw by a plain section of shaft, the twist of which is the same as that of the crank throw, if the crankshaft is loaded by moments acting at the crankshaft journals. The crankshaft, however, is loaded chiefly by forces acting at the crank pins. R. Grammel called these two conditions of loading, torsion of first and second kind. These two conditions should be distinguished, as completely different torsional deflections are obtained even with the same static moments.

An analysis of a complete crankshaft shows that the forces acting on a section of the crankshaft between two crank pins, distort not only this section, but due to the reactions acting at the bearings, all sections of the shaft are distorted. This case is similar to a beam with several supports, acted upon by a load between two supports, the whole beam being deflected.

As shown in Fig. 5, there is a main torsion and additional torsions. Thus, it is not strictly correct to represent the crankshaft by an equivalent system, consisting of a shaft with discs; a load acting at a section of such an elastic system will cause a distortion of that particular section only.

Measurements carried out on internal combustion engines agree well with the period of vibration, calculated by the method based on "torsion of the first kind." The close agreement of the natural frequencies obtained by calculation and by measurement has so far been explained by the fact that the errors made in the calculation of the torsion of the first kind and those made in the mass reduction, cancel each other accidentally. The agreement does not, however, depend on accidents, even neglecting the fact that both errors are in the same direction. but has a definite reason as revealed by A. Kimmel, who found that there is a close relationship between torsions of first and second kinds.

The "influence number" (it is the reciprocal of the torsion) of the first kind of a section of a crankshaft between two crank-pins is composed additively of the "influence numbers" of the main torsion of the second kind of this crank-shaft section and of the primary additional torsion caused by the loading of the adjacent crank-shaft section. (The other additional torsions are negligible). The reduction of this relation can be easily understood and is briefly explained below. Fig. 6 shows a 6-throw crankshaft loaded by moments M′ and M″ at its ends and by torsional forces T at the crank pins. This loading, generally occurring in torsional vibrations, can be replaced by couples acting at adjacent crank-pins. The loading by the forces T and the moments M′ and M″ is equivalent to the loading by the forces P and the moments M′ and M″ if the sum of the forces P acting at a crank-pin equals the tangential force T acting at this pin, and if the following relation for the moments at the end of the shaft is valid :—

$$M' = P_o r \text{ and } M'' = P_e r.$$

For the torsion of second kind, the following equation for the distortion of a crank-shaft K, is valid :—

$$\vartheta k\text{–}1 - \vartheta k = h_{k,k-1} M_{k-1} + h_{kk} M_k + h_{k,k+1} M_{k+1} \quad (2)$$

The torsion of first kind is obtained from the torsion of second kind by simple consideration. If in the torsion of first kind the loading by the moment M acting at the ends of the shaft, is made equal to the moment $M_k = P_k.r$ acting in the considered crankshaft section k under torsion of second kind, and if this loading is again replaced by pairs of equal forces acting at neighbouring crank-pins, then the crank-shaft section k under consideration is distorted by torsion of second kind, the following equation applying :

$$\vartheta k\text{–}1 - \vartheta k = (h_{k,k-1} + h_{kk} + h_{k,k+1}) M_k \quad (3)$$

As only one "influence number" h_k is obtained for each crankshaft section in the tests by torsion of first kind, the following relation is valid :—

$$(h_{k,k-1} + h_{kk} + h_{k,k+1}) M_k = h_k M_k$$

$$\text{or} \quad h_k = h_{kk} + h_{k,k-1} + h_{k,k+1} \quad (4)$$

i.e., the "influence number" of torsion of first kind is built up of the same "influence numbers" occurring in the torsion of second kind, viz., the "influence number" of the main torsion and the two primary additional torsions.

Comparing equations (4) and (5), the following conclusions can be arrived at. If $M_k = M_{k-1} = M_{k+1}$, then consideration by torsion of first and second kinds are equivalent. This will also be the case if the torques of adjacent crankshaft sections differ by the same amounts, and if the influence numbers of the additional torsions $h_{k,k-1}$ and $h_{k,k+1}$ are equal.

These conditions are not strictly, but very nearly, fulfilled in vibrations occurring in practice with none, or at the most one, node in the region of the crankshaft throw (Fig. 7). This is the reason why, in the case of such vibrations there is a close agreement between the calculation by torsion of first kind, and the basically correct method of torsion of second kind.

The above conditions are not valid for vibrations of higher degree, i.e., with two or more nodes in the region of the crankshaft throw. In practice, however, conditions like these have not been known so far; thus the necessity of making calculations by torsion of second kind has not yet arisen.

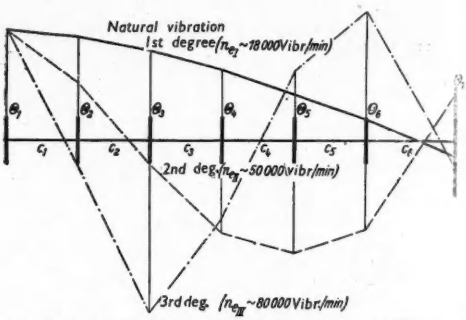


Fig. 7 Natural vibration of six cylinder engine with Flywheel

DYNAMIC EXPERIMENTS ON TORSION OF FIRST AND SECOND KIND.

In order to determine the influence of the type of loading on the natural frequency which so far has been found only experimentally, dynamic experiments were carried out on a 6-cylinder engine.

The arrangement is shown diagrammatically in Fig. 8. The engine is driven by a swinging-frame motor. At the free end of the crankshaft a vibration exciter and a DVL-torsiograph are fixed. The connecting shaft between engine and electric motor was arranged in such a way that with the vibration of second kind a node was formed roughly in the middle of the crankshaft, so getting a maximum difference between torsion of first and second kind.

The natural vibrations of first and second kind were measured by loading the engine. The torsion of first

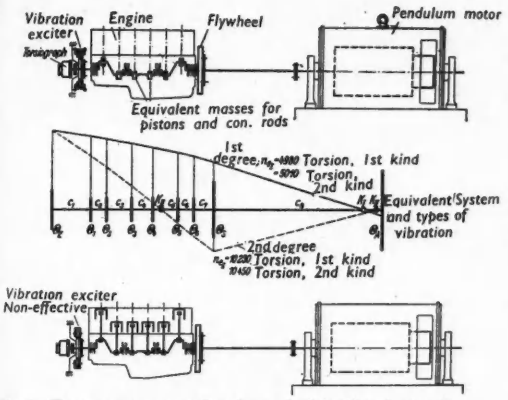


Fig. 8 **Test arrangement for dynamical loading by torsion of 1st and 2nd kind.**

kind was obtained by an exciter of torsional vibrations fixed at the free end of the crankshaft, acting in the following way :—Two small gears, on the axes of which eccentric weights were mounted, mesh with a fixed gear. The centrifugal forces produced by the rotation of these weights result in a sinusoidal torque. The pistons and the connecting rods had to be disconnected and replaced by masses at the crankpins during the measurement of the torsion of first kind, to prevent a loading by torsion of second kind. The excitation of the torsion of second kind was attained by the reciprocating masses of the piston and the connecting rod. Here the exciter of the torsional vibration was made ineffective by removing the eccentric weights.

The following results were obtained during the measurements of the natural vibration of the first and second kind (n_{eI} and n_{eII}) by means of the DVL-Torsiograph :—

Period of Vibration.	Torsion of 1st Kind.	Torsion of 2nd Kind.	Error %
n_{eI}	4,980	5,010	—0.6
n_{eII}	10,230	10,450	—2.1

Thus the experiments agree with the results obtained by calculation, the natural frequency of a definite system depending on the kind of loading. If, however, in the case of vibrations occurring in practice resulting in one or only one node in the region of the crankshaft throw, the vibrations of the second kind are not taken into account, only a negligible error being produced.

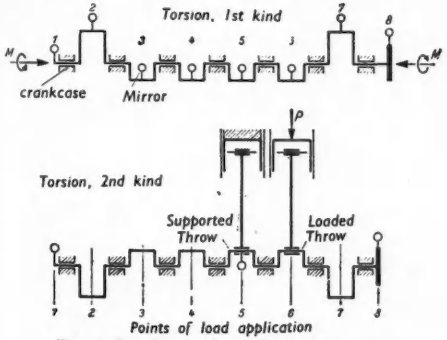


Fig. 9 **Loading in the static torsion test.**

THE DETERMINATION OF THE TORSIONAL STIFFNESS.

It is possible in principle to calculate the main and additional stiffnesses, assuming the crankshaft consists of thin rods, and the forces in the crankpins and crankshaft-journals are acting at points or else their distribution along the journals is well defined. This assumption was sometimes permissible as a first approximation, with the crankshafts of the older low-speed engines ; in modern engines, however, this assumption is not justified. The webs of the modern crankshaft are shaped more like discs than rods, and the crankpins and main journals usually overlap. Thus, it is impossible to determine the torsional rigidity mathematically, and it is necessary to determine it by means of empirical formulæ.

EXPERIMENTAL DETERMINATION OF THE TORSIONAL RIGIDITY.

Static Torsion Tests.—The determination of the torsional rigidity by means of static torsion tests by torsion of first kind is known. The crankshaft mounted in its crankcase is loaded and unloaded by torques applied at its ends. The deflections are measured optically by means of mirrors fixed to the crankshaft throws.

In the determination of the torsional rigidity by torsion of second kind, the crankshaft mounted in the crankcase is loaded by two forces acting at two crankpins respectively (Fig. 9, bottom). In R. Grammel's

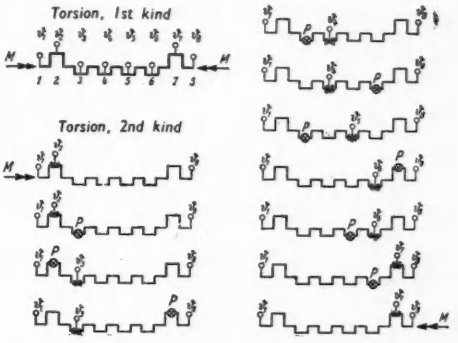


Fig. 10 **Experiments for determination of the influence numbers of a six throw crankshaft.**

method one piston is loaded and unloaded in steps by means of oil pressure in the cylinder, the other piston being kept in position by means of a distance piece fixed between the piston and the cylinder head. Instead of using a hydraulic system, weights can be used to apply a load. The distortion of the crankshaft is measured at both ends and at the fixed throw of the crankshaft by means of mirrors and telescopes.

The friction in the bearings and also in the cylinders results in an increased hysteresis effect, causing great difficulties in the evaluation of the results. This difficulty can be partly overcome by making the loaded crankshaft vibrate by means of a vibration exciter.

While it is necessary to determine only one torsional rigidity for each throw and both crankshaft ends in the case of torsion of first kind, in the case of torsion of second kind at least three values (main torsion and two additional torsions) have to be determined for each crankshaft section.

DYNAMIC TESTS.

Instead of the static torsion test, dynamic measurements on the stationary crankshaft can be carried out in order to determine the torsional rigidity. A diagram

of the required arrangement is shown in Fig. 11. The crankshaft is mounted as under operating conditions and is loaded by weights equivalent to the pistons and conrod, the flywheel being in position as well. The crankshaft is made to oscillate by means of a centrifugal exciter, resonance conditions being obtained by altering the speed of the eccentric masses. The amplitudes of the vibrations are measured by mirrors mounted as shown in Fig. 11. The natural frequency is obtained directly from the speed of the driving motor. If the mass of the centrifugal exciters is made equal to the masses mounted on the free end of the motor, the test gives the natural frequency of the actual driving systems.

Knowing the natural frequency, the amplitudes and the moments of inertia, the torsional rigidities of the different crankshaft sections can be obtained by elementary vibration calculation. The test is relatively simple to carry out, and gives quite reliable results for the torsional rigidity by torsion of first kind.

It is known that the natural frequency can also be determined by torsiographic measurements on the running engine. The arrangement used is shown in Fig. 12. To calculate backwards by the usual method, the torsional rigidity from the measured natural frequency, and the known moments of inertia, is only correct in the case of homogeneous motors, i.e., motors in which the throws, connecting rods and pistons are all equal. As modern engines, however, are usually non-

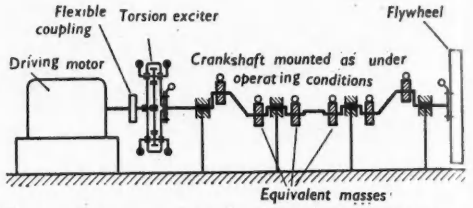


Fig. 11 Test arrangement for dynamic measurements on the non-rotating crankshaft.

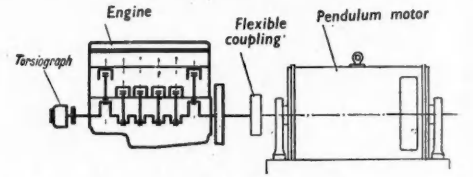


Fig. 12 Test arrangement for torsiographic measurements on the engine.

homogeneous, the torsional rigidities of the separate throws are better determined from the dynamic experiments on a non-rotating crankshaft.

NEW OIL-BLAST BREAKER FOR MEDIUM VOLTAGE.

By A. ROTH. (From *Bulletin Assoc. Suisse des Electriciens*, Vol. 34, No. 10, May 19th, 1943, pp. 291-294).

As recently as 15 years ago, the circuit breaker was still looked upon as a weak point in networks of considerable short circuit capacity. Since then the position has been remedied by the introduction of breakers with small oil content, and of compressed air breakers. It is a singular fact that in Switzerland the time-honoured oil circuit breaker has maintained its place up to the present day in small plants operating at voltages up to 30 kV. But this is probably due to the fact that compressed air circuit breakers had to be ruled out because of the expense involved in providing the necessary compressed air plant, which, incidentally, must be carefully looked after. Or this negative attitude may also have been due to the fact that the interrupting capacity of this type lies below that of 200 MVA (at 8 kV) required in Swiss practice.

This situation suggested the development of an oil-blast breaker for lower voltages, its design to be based on the 45 kV indoor breaker of this type, which, because of its simplicity of design and service reliability, has been finding ever-increasing popularity. In the course of investigation it was soon found that this breaker principle offers several additional advantages. Outstanding among these is the possibility afforded of installing the switching plant on the single-level system, hereby achieving a particularly simple and neat circuit arrangement combined with small space require-

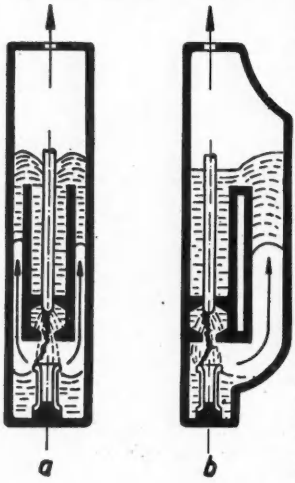


Fig. 1a. Flow path of gas and oil in old type switch.

Fig. 1b. Flow path of gas and oil in new type switch

ments. The foolproof features of this switch type are another advantage. Also, because of the small oil contents, the danger of oil fires is minimised. Finally, the high interrupting speed, secured by the most simple means, is also an important point in favour of this switch type, which point should not, however, be over emphasized.

The chief difficulty in the way of designing of oil switches with small oil content for voltages below 45 kV, and especially so below 30 kV, is the dimensions of the metal flanges placed around the cylindrical explosion chambers made of laminated bakelite. These flange dimensions lead to an excessively great width of the switch and to a correspondingly great cell width. This could, of course, be avoided with the use of moulded plastic materials or of porcelain, as these materials can be formed into shapes other than a cylinder. There are, however, certain objections to the us of these materials.

In the present design, this difficulty has been eliminated by a radical change in the shape of the oil-

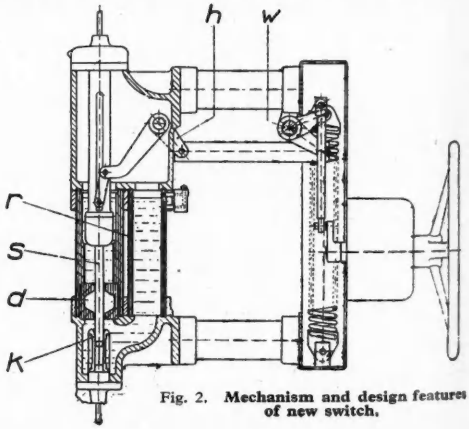


Fig. 2. Mechanism and design features of new switch.

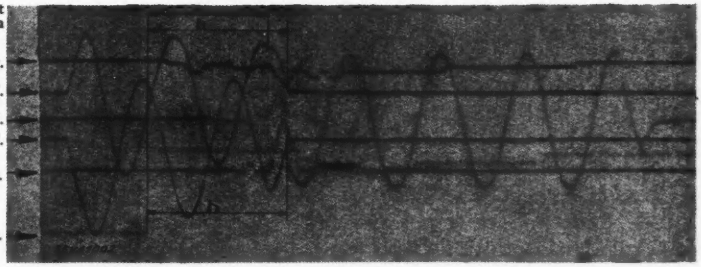


Fig. 3. **Oscillogram with breaking at 5,800 A and 23 kV., with oil switch designed for 20 kV, 600 A.**

blast switch as such. While in the design heretofore used (Fig. 1a), the gases escape through the annular space between explosion chamber and the inner vessel wall, the outer oil vessel and the explosion chamber are amalgamated as shown in Fig. 1b. However, a second tube is provided for the escape of the gases. This design also secures a diminution of the quantity of oil required, since the oil-filled space above the explosion chamber must naturally decrease with the diameter.

In tripping the switch, the compressing spring held by the locking device of the handweeel is made free to turn the shaft w, the rotation of which is transmitted to the lever k (Fig. 2). The latter, therefore, by means of an interposed linkage, pulls upward the contact rod s with great velocity. This causes an arc to form between the rod and the stationary plug contact k which is seen to be of a flared design. The nascent gas will, therefore, displace a certain amount of oil which, as is well known, will lead to an upward movement of the oil within the pipe r. As the latter pipe is made of ample size, gas pressure will be comparatively small ; and this is reflected by a correspondingly small voltage drop Δu and liberated energy $i\ \Delta u$ (kW).

After approximately 0.01 seconds, the lower end of the contact rod s—and with it the topmost portion of the arc—enter into the explosion chamber d. This causes a high gas pressure to be set up in the explosion chamber, particularly since, during the peak period of the a.c. sinusoidal curve of the current, the ions and electrons of the arc are blocking escape of the oil through the lower orifice. But as soon as the current passes through zero, the orifice is rendered unobstructed so that a jet of oil and gas can pass through the orifice and cool the arc and extinguish it. An oscillogram taken at 23 kV and 5,800 A and reproduced in Fig. 3 shows that the arcing time is no more than 3/100 of a second.

Fig. 4. **View of 20kV, 600 A switch of 200 MVA capacity**

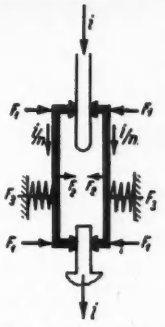


Fig. 5. **Forces acting upon stationary contact segments.**

Apart from the simple and neat electrical operating characteristics of the switch, its simple mechanical features are also worthy of note. It is seen that its mechanism is no more complicated than that of the common type of oil circuit breaker ; as a matter of fact it lacks the cross-head employer with the latter type. This is a point which greatly facilitates neatness and machine-like design of the mechanism. Normally, the switch is equipped with the usual hand-wheel and trip-free mechanism (see Fig. 4), but remote control with a spring drive can also be incorporated. Tripping because of over-current or short circuit is effected with the use of an overload relay which may be of the simple overload or of the definite-time variety, the tripping impulse being mechanically transmitted by means of a link and lever gear made of insulating material. An independent source of electric power supply is not therefore required. This enables the use of any type of secondary relays in connecting with voltage or current measuring transformers.

Referring to Fig. 5, it is seen that the contacts are fully balanced in electro-dynamic respects. The separating forces F_1 occurring during the passage of the current through the contact points are completely balanced by the branch currents i/n of the contracting forces F_2 of n individual flared segments ; and this

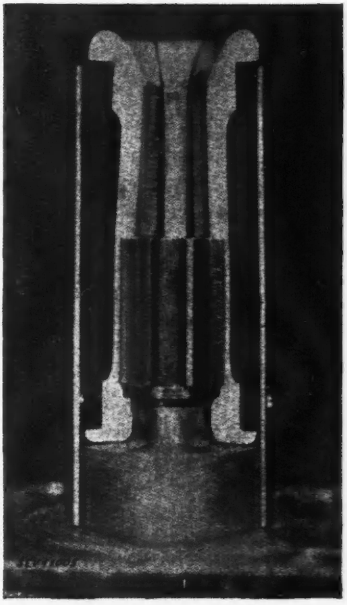


Fig. 6. **Electrodynamically balanced stationary contact after closing twice at 50,000 A peak value.**

is quite independent of the current intensity because both F_1 and F_2 are porportioned with i^2/n. This effect was achieved by approplate dimensioning of segment-length and diameter of the contact rod. The only remaining force is, therefore, the pressure F_3 exerted by the springs which are adjusted to give the required contact pressure. The surprising effect obtained with this measure can be gauged from the photograph reproduced in Fig. 6, which shows the condition of a contact after having closed a current of 50,000 A max. amplitude.

It may be added that the contact points proper are plated with silver in order to prevent an increase in contact resistance by copper oxidation. Owing to the balanced design of the switch, the force required to close it is in no way increased by electrodynamic forces even under short-circuit conditions. This greatly facilitates manual operation, and also decreases the power requirements in the case of spring motor operation.

The switch shown in Fig. 6 is designed for 20 kV and 600 A, interrupting capacity being 5,800 A at 20 kV and 11,500 A at 10 kV. When closing the switch, a peak value of 38,000 A is permissible. Interrupting time reckoned from the time of tripping to the extinction of the arc is 0.05 seconds. With a spring drive, closing time is 0.2 seconds. The oil contents are 3.2 litres, and the quantity of gas developed when breaking 5,800 A at 23 kV amounts to about 20 litres per phase (this referred to atmospheric pressure) approximately 100 kilojoules being set free per phase.

THE ERICSSON AUTOMATIC TELEPHONE SYSTEM.

By Dipl. Ing. H. SPÜLBECK. (From *Ericsson Review*, Vol. 1943, No. 1, pp. 12-15).

THE use of 500-line selectors in large Ericsson systems is based on the following considerations. Traffic statistics show that, as a rule, of one hundred subscribers no more than ten make a call at the same time. It is therefore sufficient to instal ten final selectors for every one hundred subscribers. Considering, however, the traffic statistics covering five hundred subscribers, one will find that the volume of calls is much more uniform than in the case of one hundred subscribers. For this reason, the same service quality can be achieved by employing only 6 per cent of final selectors. This means that 20 selectors less are required for the 500-line system as compared with the 100-line system.

The Ericsson 500-line system is motor driven. In Fig. 1 is shown a part of an incompletely assembled selector, only the last 14 frames being in place. The wiper arm is seen to stand at the 24th frame. In Fig. 2, it is seen that a large number of selectors can be arranged in a rack, the selectors being stacked one upon the other. As each selector is only 35 mm. in height, one rack can accomodate as many as 40-70 selectors. With 50 selectors placed in one rack, $50 \times 500 = 25{,}000$ different connections can be made. Yet the multiple fields have only 500 soldered joints. This represents an important advantage of the Ericsson system.

In the Ericsson system, identical selectors are used for line-finders, group selectors and final selectors. The basic circuit arrangement of an Ericsson exchange is shown in Fig. 3. Because of the non-decimal construction of the system, the motor-driven selectors require registers for receiving the dialling impulses of the subscriber and for translating them. According to Fig. 3, there are three ways in which a register can be brought into play. (1) Over a register finder, (2) over a register selector, and (3) over an allotter. When a subscriber lifts the receiver, a start distributor puts six to eight line-finders into action in order to locate the calling party. When one of the line-finders has found the multiple frame, the others are stopped, while the successful line-finder radially extends its contact arm over the multiple frames until contact is made with the calling line. At the same time, a free register is connected to the line-finder involved. It is only then that the subscriber hears the dialling tone and can begin to dial. The number dialled by the calling party is accepted by the register, and if it is a five-digit number, for instance, it is translated into a three-digit number. The group selector connected to the line-finders now begins its rotary movement, sending for each step a revertive impulse to the register. When the group selector has made the requisite number of steps and has sent back the requisite number of impulses to the register (corresponding to the transcribed number), it is stopped by the register.

The group selector now changes its contact arm from rotation to radial movement over the multiple frames,

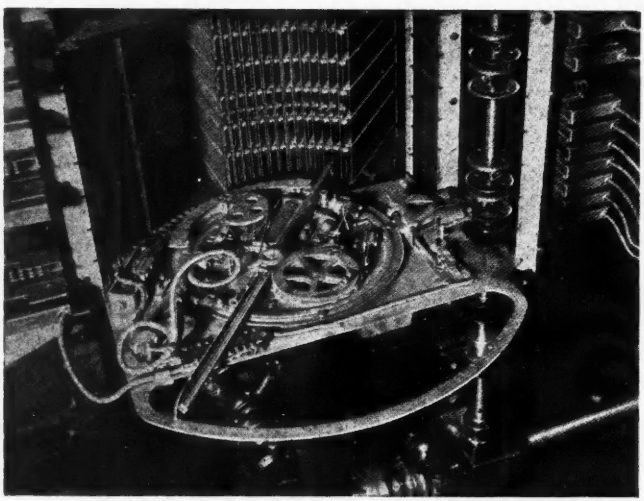


Fig. 1. **Ericsson 500-line selector.**

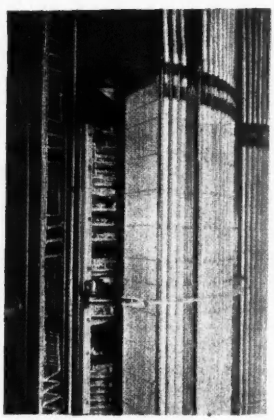


Fig. 2.
Selectors installed in rack. Rear view.

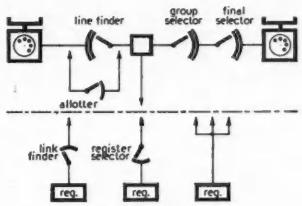


Fig. 3. **Ericsson circuit.**

thus searching for a free final selector. Both kinds of movement of the latter are controlled with the help of the back impulses of the register, and finally the required outgoing line is reached. The Ericsson system is said to offer the advantage of requiring only one type of selectors. But, as will be seen from Fig. 3, other selector types are also employed, as e.g., uni-selectors, which serve as start-distributors, as register selectors, or as outgoing and revertive impulse selectors in the register. Selectors of a special type are shown in Fig. 4. These barrel-type selectors, with their banks extending through 360 degrees, are employed for selection of a free register. They are, however, used only in older installations, as in modern installations their place is taken by the uni-selector type shown in Fig. 5, which are of the self winding clock-work driven type.

The selectors arranged in the racks are driven by a shunt type motor of 1/8th h.p. The horizontal main

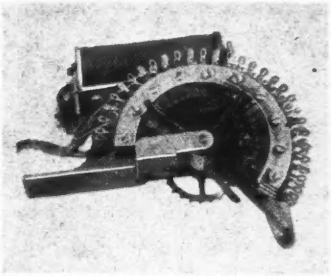


Fig. 5. **Uni-selector.**

drives are coupled to the vertical drive-shafts of the individual selector panels, the vertical shafts revolving at 48 r.p.m. The contact arm of the 500-line selector passes over the contact bank from one home end to the other in 1.75 seconds, while in its radial movement it covers 20 multiple frames in one second.

An exchange for 500 subscribers is shown in Fig. 6. For the supervision of the register selectors and relays, a control board is provided which has a white light for each forward impulse selector and a red light for each revertive impulse selector. Burning of a lamp indicates that the corresponding selector is engaged. Burning and extinguishing of the lamps in the established sequence therefore serves as proof that the exchange is functioning properly.

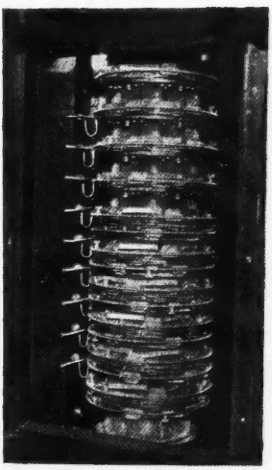


Fig. 4. **Barrel-type register finder.**

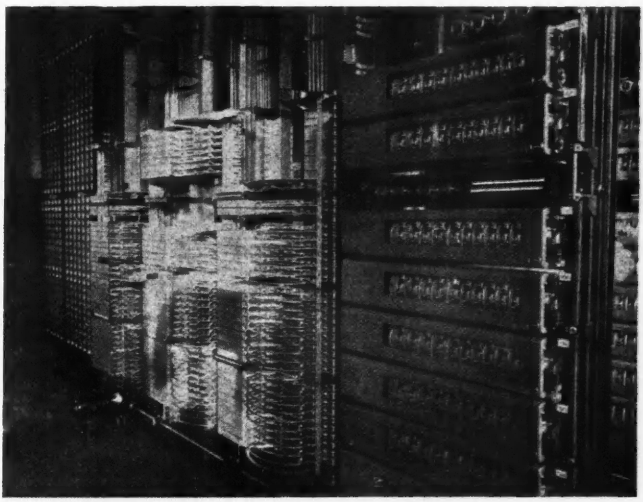


Fig. 6. **500-line Ericsson exchange.**

THE LAURETTE OIL FILTER.

By H. PETIT. (From *La Vie Automobile*, Paris, Vol. 39, Nos. 1259-1260, September 10th and 25th, 1943, pp. 230-231.)

REGENERATION of the lubricating oil, a long neglected economic measure, has to-day become a matter of absolute necessity. While formerly oil suppliers were wont to recommend that the oil contents of the crank case should be drained off and replenished every 2,000–2,500 kilometers, such a procedure to-day could only be considered an intolerable luxury. But this means that as a sheer matter of prudence oil filtration and regeneration plant should be installed if serious accidents are to be avoided.

Various devices have been prepared or built for this purpose, as, for instance, the Lautrette-L.C. filter. According to the claims of its designer, this apparatus combines continuous oil filtration with the elimination of the light volatile constituents of the oil and enables re-introduction of the latter constituents in vapour-phase into the cylinders where they play a useful role in the lubrication of the pistons.

The filtration proper is effected in a cartridge containing Fuller's earth, the porosity of which is arranged

to increase from the outer surface of the annular cartridge towards the interior. This cartridge serves to retain and eliminate all solid particles or impurities that may be present in state of suspension in the crankcase oil. Distillation of the oil takes place in the central portion of the filter in a heat-exchanger device which receives its heat supply from the exhaust gases. This distillation, which takes place under slightly sub-atmospheric pressure and which proceeds while the oil flows slowly along the exhaust-heated central tube, is not, however, accompanied by cracking, but leads to the entrainment of the volatiles contained in the oil. It is this combination of filtration and distillation which constitutes a true process of oil regeneration. On the other hand, the fact that the distilled-off volatile constituents are passed into the cylinder via the admission valve, creating the aforementioned sub-atmospheric pressure in the filter, is of considerable importance, as the vapours of the light oil fractions are known to be particularly beneficial to cylinder lubrication.

Referring to Fig. 2, the filter cartridge F is seen to be arranged in a sheet metal cylinder A. As already mentioned, the Fuller's earth contents of this cartridge are of varying porosity, being composed of two layers of different density. The layer of greatest density is near the surface, and that of smaller porosity near the centre, in accordance with the decreasing cylindrical flow area towards the centre. The purpose of this measure is, of course, to equalise flow resistance through the filter as much as possible. The filter as such is comprised in a perforated cylindrical sheet-iron container, which is held in place by means of the helical spring D and the washer C, which serve to press the upper end of the cartridge against the gasket B interposed between cartridge and top part of the filter casing A. As Fig. 2 shows, the centre part of the filter is formed by a corrugated tube which forms the last oil pass, while its outside is swept by the exhaust gases admitted at the bottom through the openings G. The gas flow is guided by a concentric baffle tube which is seen to be open at its upper end, thus enforcing reversal of the gas flow at that point. At the bottom of the corrugated tube K the outlet H is provided, from whence the altered oil is returned to the crank case, while the contaminated oil is admitted to the filter through inlet E. The upper end of the corrugated tube is seen to be connected to a cylindrical part from which the volatile constituents are withdrawn through the opening N communicating with the ejector P, the outlet of which leads to the admission line of the engine.

The flow through the filter is obtained by tapping the lubrication circuit of the engine. Entering at E, the contaminated oil flows into the annular space between filter casing and outer surface of the filter cartridge. After deposition of the solid impurities upon the external surface of the cartridge, the oil passes through the filter filling, and then passes in upward flow through the orifices M into the cylindrical top part of the corrugated tube, in which it descends to the outlet H at the bottom.

Referring to Fig. 2, it will be clearly understood that the oil and exhaust gas passages constitute a counter-flow heat exchanger, the hottest gases entering at the point where the purified oil passes out from the filter. This arrangement serves to ensure that the oil temperature is elevated to a level at which both its water contents and the light hydrocarbon fractions are distilled off, this distillation taking place under the partial vacuum created by the aforementioned ejector P.

The filter should be installed at the hottest point under the engine hood, possibly in the proximity of the engine exhaust. The oil temperature in the corrugated tube should be adjusted to 100–120° C. This temperature range will give the desired distillation effect without leading to cracking of the oil. Prior to its return to the crank-case, the oil is led through a pipe cooled by the air stream of the engine fan. In cases where the oil pressure is less than 2 kg/sq. in., a suction pump should preferably be installed with the filter. Filter tests conducted at the Bellevue national laboratory and also at another laboratory have proven the efficiency of the filter device. After 16 hours of operation, the flash point of the oil was found to be reduced from 221° C. to 207° C., which shows that the hydrocarburets were completely eliminated.

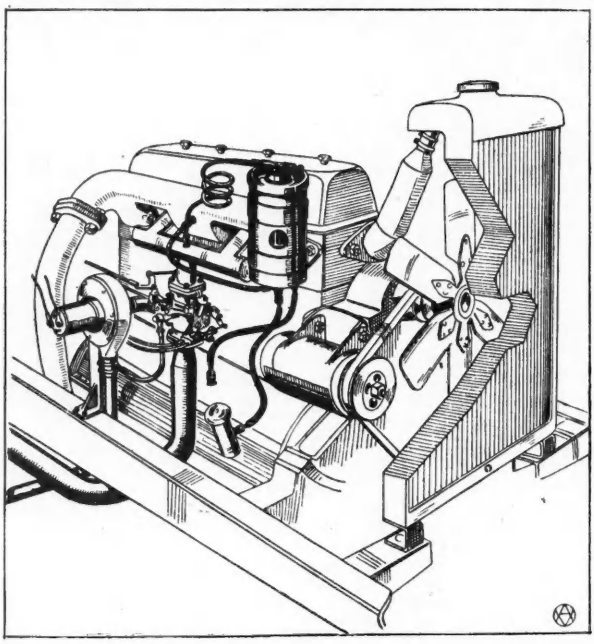


Fig. 1

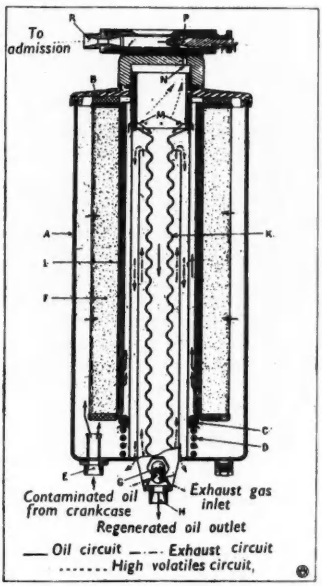


Fig. 2

STRENGTH OF ARC WELDED SEAMS.

By H. CORNELIUS and F. BOLLENRATH. (From *Luftfahrt-Forschung*, Vol. 20, No. 6, 30th June, 1943, pp. 175-180.)

IN aircraft production, metallic arc welding has been making good progress over a number of years, partly because of its superior speed as compared to oxy-acetylene welding, and partly because of the smaller distortion produced. In the course of this progress, a number of tests were made to study the properties of butt welds produced by metallic arc welding of high strength, heat-treated steel sheets of 3, 4 and 6 mm. thickness of the following composition :

Material	C %	Si %	Mn %	P %	S %	Cr %	Mo %	V %	Ni %
Steel 1.. ..	0.26	0.38	1.25	0.021	0.011	0.85	*	0.17	*
Steel 2.. ..	0.26	0.40	0.70	0.011	0.013	1.07	0.22	*	*
Steel 3.. ..	0.29	0.25	0.75	0.023	0.023	2.19	*	0.21	*
Core of Electrode, 4 mm. dia.	0.15	0.22	0.68	*	*	0.62	0.22	*	0.25
3-mm. Electrode Welding material ..	0.13	0.05	0.68	*	*	0.35	0.21	*	*
4-mm. Electrode Welding Material ..	0.14	0.05	0.67	*	*	0.38	0.21	*	0.22

* Not determined, only traces, or not at all present.

The joints were made with coated electrodes of 3 and 4 mm. core diameter respectively, the core containing some chromium and molybdenum, and also a small amount of nickel originating from the scrap used in the production of the material. Considering that in aircraft construction, welding can often be carried out from one side only, V-type test welds were made. In order to minimise distortion and warping, the cold sheets were clamped in a welding jig which did not, however, impede freedom of shrinkage. The length of each test weld was approximately 200 mm., and the V angle 70, 80 and 90° for sheet thicknesses of 3, 4 and 6 mm. respectively. The finished welds were normalised for 30 mins., and then heat-treated by quenching in oil and subsequent tempering for one hour at 400, 500, 575 and 650° C. respectively.

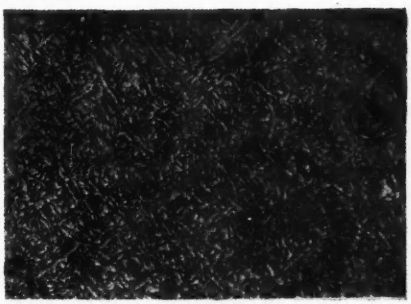


Fig. 1. **Microstructure of welded joint of sheet of 4 mm. steel No. 1, not heat-treated after welding.**

Metallographic examination of weld seams not heat-treated after welding could naturally yield no definite information regarding the alloying of the seam by the alloying constituents of the sheet itself. As Fig. 1 shows, the seams of the sheets of No. 1 and No. 2 material possess in the main a fine Widmannstätten structure ; while, according to Fig. 2, the seams of the No. 3 steel sheet have a preponderantly coarser structure, tending less to acicular ferrite formation, but showing a quantitatively more pronounced eutectoid constituent. Consequently, the hardness of the seam of the No. 3 steel sheet has proved to be greater than that of either No. 1 or No. 2. From this, it can be concluded that the alloying action of the No. 3 material upon the weld is greater than that of the other two materials.

With increasing sheet thickness, the hardness of the seam decreases ; and with steels No. 1 and No. 2, the maximum hardness also decreases in the vicinity of the weld as far as it is affected by the heat of the welding process. This is due to the diminution of cooling speed with increased thickness of the sheet. However, with steel No. 3, maximum hardness in the proximity of the weld is independent of the thickness of the sheet ; while in the case of steel No. 1, a sheet thickness of 3 mm. is the limit for martensite formation in the vicinity of the weld.

The results of tensile tests conducted on both welded and unwelded pieces of 3 and 4 mm. sheet thickness are charted in Figs. 3–5. The elongation values are included in order to indicate the location of the fracture relative to the seam, an elongation considerably lower than that of the original sheet indicating the occurrence of fracture either in the weld itself or in

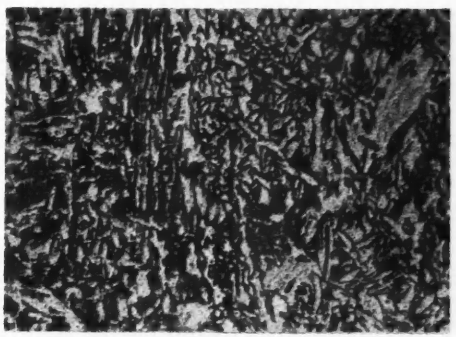


Fig. 2. **Microstructure of welded joint of sheet of No. 3 steel not heat-treated after welding.**

the transition zone. Vice versa, high values of elongation correspond to fracture in the parent metal at some distance from the weld. The following conclusions can be drawn : Under conditions of heat treatment leading to tensile strength values exceeding 130 kg/sq. mm. of the sheet, the weld seams of 3 mm. sheets attain the strength of the original parent material in some cases, but there is an unduly large scatter towards lower strength values. Where the welded seam was not ground down, fracture occurred mostly in the transition zone between weld and parent metal and rarely in the latter ; but where it was ground down, fracture took place mostly in the seam, rarely in the transition zone, and only exceptionally in the sheet. Thus, in spite of the alloying action to which the welded material is subjected from the side of the parent metal, strength of the joint benefits less from heat treatment than the material of the sheet. But this matters little with 3 mm. sheets heat-treated to 130 kg/sq. mm. strength, and with the welded seam left in place. With heat treatment adjusted to produce a strength of 90 to 120 kg/sq. mm. in the case of steels 1 and 2, and of 90–130 kg/sq. mm. with steel 3, the respective strength of the joint is approximately equal to that of the material. Steels with less than 120 kg/sq. mm. strength, almost without exception suffer fracture in the sheet itself at some distance from the seam if the latter is left untouched ; but where the seam is ground down, fracture takes place in the joint with tensile strength values as low as 90 kg/sq. mm. Thus, while the welding electrode used

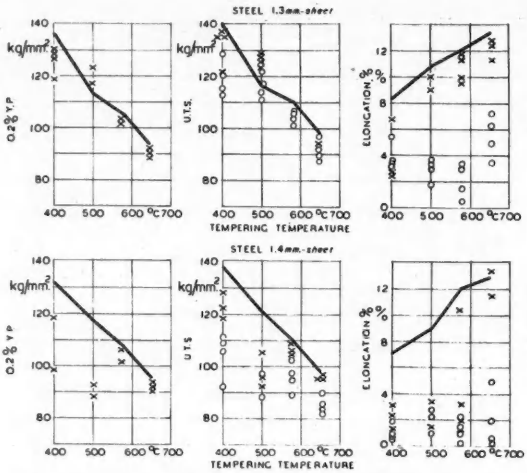


Fig. 3. **Strength of original and of welded sheets of No. 1 steel in heat treated state. Curves refer to original sheet ; " o " to welded sheet with weld ground down ; " x " to welded sheet with weld in original state.**

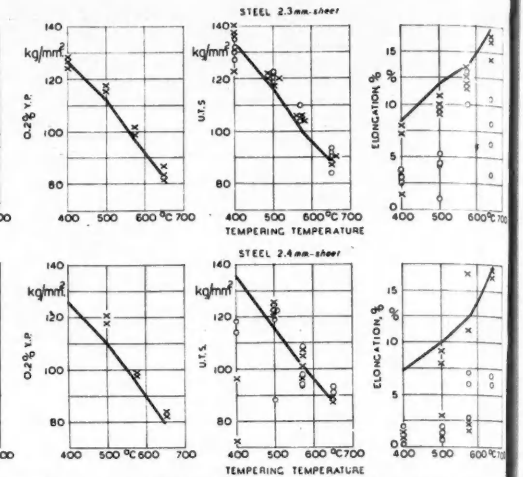


Fig. 4. **Strength of original and of welded sheets of No. 2 steel in heat treated state. Curves refer to original sheet ; " o " to welded sheet with weld ground down ; "x" to welded sheet with weld in original state.**

in the test is suitable for welds of high static strength when applied to 3 mm. sheets, its applicability to 4 mm. sheets is limited to medium strength values. With a tensile strength from 90–100 kg/sq. mm. and with the welding ridge left untouched, fracture cannot always be expected to take place in the base material. On the other hand, heat treated steel 2 exceeded 120 kg/sq. mm. strength with the welded seam left in place, the fracture occurs at a satisfactory distance. The applicability of the low-alloy electrode, however, ceases in the case of 6 mm. sheet heat treated to high tensile strength.

Considering the results of the static tensile test, fatigue tests could have been limited to 3 mm. sheets. But it had to be considered that dynamically stressed, heat treated welded joints of high strength quite frequently possess a smaller static strength than the sheet itself, as, e.g., in the case of American-built hollow steel airscrews. For this reason, fatigue testing was extended to both 4 mm. sheet of average tensile strength and 4 mm. sheet in which the tensile strength of the weld was below the high tensile strength of the sheet itself.

Fatigue testing of the 3 and 4 mm. materials was carried out with a direct stress fatigue machine of the *Schenk-Erlinger* type*. The findings were summarised as follows :—For 10^4 load cycles, the semi-range of stress of the heat treated welded joints of the 3 mm. sheets of the three materials lies between 42 to 58 kg/sq. mm. With 10^4 cycles, no definite influence either of the composition of the material or of the static tensile strength of the weld upon the semi-range of stress could be observed. But ground-down welded joints of the 4 mm. sheet with the weld strength lying below that of the sheet, exhibited a smaller semi-range of stress than untouched welded joints with a weld strength corresponding to that of the sheet. With regard to semi-range of stress, untouched joints of 4 mm. sheet correspond to both untouched and ground-down seams of the 3 mm. material. No influence of the composition of the steel upon the semi-range of stress could be found, nor did the untouched welds of the 4 mm. material show any interrelationship between tensile strength of the weld (within the range of 80-120 kg/sq. mm. investigated) and the semi-range of stress based on 10^4 cycles.

* In this machine specimens can be pulled out and pushed back by rapidly applied tensile and compressive stresses, without showing either permanent deformation or failure from fatigue.

Similarly to the semi-range of stress, the endurance limit (based on 10^7 load cycles) of the welds of the 3 mm. material, proved to be uninfluenced by either the composition of the material or by the weld strength when the latter ranges from 100–130 kg/sq. mm. Unlike the semi-range of stress, the endurance limit was found to vary according to whether the seam was ground down or not. With the seam left untouched, the initial range of stress was evaluated as 18–23 kg/sq. mm., but this rises to 30–49 kg/sq. mm. if the seam is ground down. Similar conditions prevail with the material of 4 mm. thickness, although the improvement obtained by

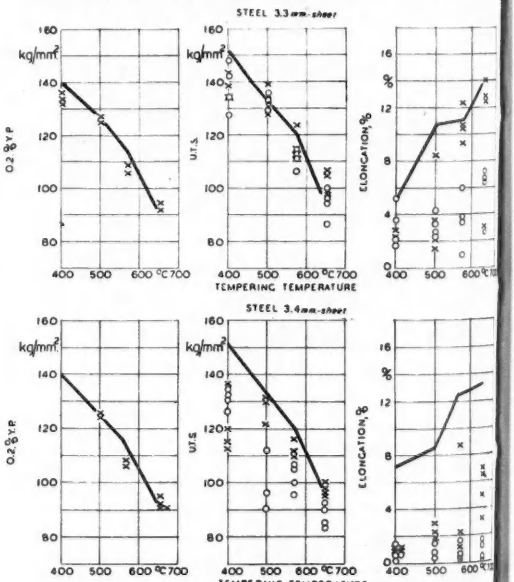


Fig. 5. **Strength of original and of welded sheets of No. 3 steel in heat treated state. Curves refer to original sheet ; " o " to welded sheet with weld ground down ; " x " to welded sheet with weld in original state.**

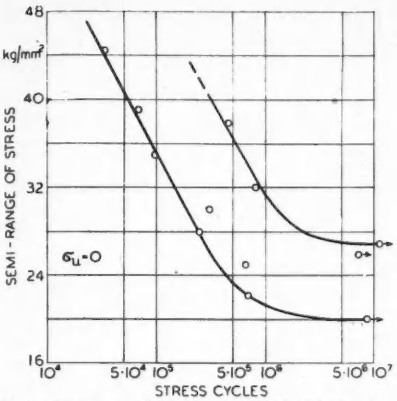


Fig. 6. **Fatigue test values of welded joints in heat treated 3 mm. sheet of No. 1 steel of 100-110 kg. per sq. mm. tensile strength with weld as made.**

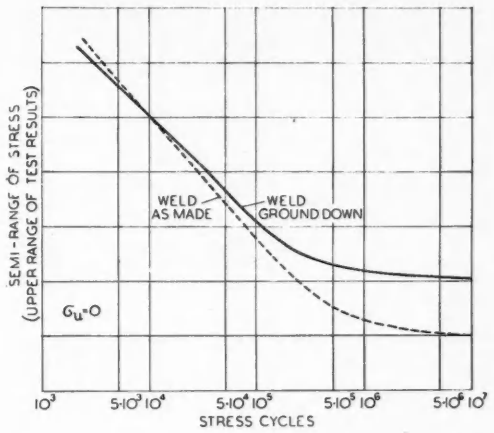


Fig. 7. **Schematic representation of fatigue test values found with welded joints of heat treated 3 mm. sheets of 100-130 kg. per sq. mm. tensile strength.**

dressing the weld is less pronounced. This may have been due to the fact that the dressed welds of the 4 mm. sheet were of smaller static strength than the material itself, or the execution of the welds may have been less perfect.

The aforementioned semi-range of stress and endurance limit values are based on the lower limit values obtained in the various endurance tests. The extent of scatter experienced is indicated in Fig. 6, which refers to untouched welds of 3 mm. sheet heat treated to 100–110 kg/sq. mm. tensile strength. From a comparison of the respective S/N curves of untouched and dressed weld seams of 3 mm. heat treated material of high strength, it is found that the curves obtaining with untouched seams exhibit greater steepness. Considering that the untouched and the dressed seams exhibit an identical semi-range of stress at 10^4 cycles, and that the endurance limit is doubled by dressing the seam, an ideal S/N curve can be established, as shown in Fig. 7. It is seen from this chart that welded joints subjected to fatigue below the 10^4 cycle range need not, and should not, be dressed; while for operation above this range, dressing of the weld will greatly improve fatigue strength of the joint.

THE FRENCH GAS PRODUCER INDUSTRY IN 1943.

By J. DELPEYROUX, Director of Research, Automobile Club of France. (From *La Technique Moderne*, Vol. 35, Nos. 15 and 16, 1st and 15th August, 1943, pp. 111–116.)

DURING 1940 and 1941, the conversion of road vehicles in France to producer gas operation proceeded apace, the number of vehicles converted totalling some 50,000. This development was, however, accompanied by an enormous increase in the number of types of producer gas plants placed on the market, so that by the end of 1941 no fewer than 450 different designs could be counted. In recognition of this undesirable trend, plans for the reorganisation of the producer gas generator industry were taken in hand by the Ministry of Industrial Production. These plans, aiming at a reduction in the number of types produced, were discussed in a series of conferences held on the occasion of the Paris Fair in September, 1941, the outcome of the conference being the creation of a special Commission to deal with the technical aspects of Producer Gas Technique. By June 1, 1942, the production programme of the industry was standardised on the basis of 29 different types of gas producers, of which 15 designs concerned the use of wood fuel, the balance relying upon coal as fuel, while employment of charcoal was completely eliminated in view of the shortage of supplies. On the basis of this standardisation programme, the entire field of design was classified in ten groups as established by the Producer Gas Committee, which, on January 1, 1943, replaced the aforementioned Commission. Each one of these ten groups represents a voluntary organisation of manufacturers devoted to the development of a standard design for the class of apparatus concerned. Once these standard designs have been decided upon, no other designs will be allowed to be produced.

Acceptance of a standard design is made conditional upon the successful conclusion of a number of serviceability tests which are conducted under a variety of conditions. So far 9 types for wood fuel and 6 types for coal fuel have been accepted. In the first group, of which Brandt and Panhard are in charge, the design for wood fuel given in Fig. 1 has been adopted as standard.

This gas producer plant is seen to comprise the gas producer proper, an ejector device for slow running, a

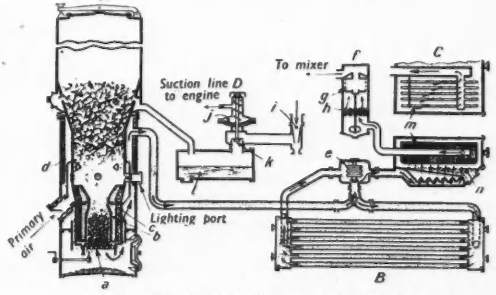


Fig. 1. **Standard Design No. 1 for Wood Fuel.**

(*A*) Producer, (*B*) Cooler, (*C*) Cleaner, (*D*) Vent.
(a) Movable grate, (b) Refractory furnace, (c) Refractory cement, (d) Glass wool insulation, (e) Thermostat-controlled by-pass, (f) Condenser, (g) Safety filter, (h) Glass wool, (i) Exhaust gas-operated ejector, (j) Rubber diaphragm, (k) Valve operated by engine throttle, (l) Condensate, (m) Paper filter elements, (n) Cork powder.

cooler with by-pass device, and a gas scrubber. Provision is made for the use of a second cooler. Referring to Fig. 1, the generator is seen to be of the down-draught type equipped with multiple tuyeres. Ample fuel capacity is secured by conical enlargement of the upper part, while the refractory furnace in the lower part is set over a moveable grate. The gas withdrawn at the bottom is passed in upward flow through an annular space formed by outer furnace wall and a specially provided surrounding sheet metal cylinder, as shown, hereby ensuring a sufficiently high temperature of the fuel. Air admitting tuyeres are arranged around the furnace at two levels after it has been preheated in a collector placed in the annular space which is swept by the hot gases leaving the furnace. The hopper is made of a single thickness of sheet, and is equipped at its base with a water trap which collects the products of the distillation of the wood as they trickle down the hopper wall on which they condense. The water trap discharges its overflow into the container *l*, while the vapour of the latter also communicates with the ejector *i* via a diaphragm operated vent valve *k*. This valve is opened by the increased vacuum in the admission system when the engine is idling, hereby venting the vapours from the hopper via tank *l*, valve *k*, and ejector *i* to atmosphere. The cooler *B* is seen to be equipped with the thermostat operated by-pass *C* which holds the gas temperature at the best suitable temperature for subsequent cleaning in the scrubber. The latter comprises a section containing cork powder and another equipped with special filter paper.

From the filter the gases are passed on to the condenser *f*, which also contains a glass silk filter *h* and a safety filter *g* prior to the discharge of the gas to the mixer.

The wood producer gas unit which constitutes standard model No. 2 (built by the companies La Lilloise, Gohin, and Unic) is shown in Fig. 2. This

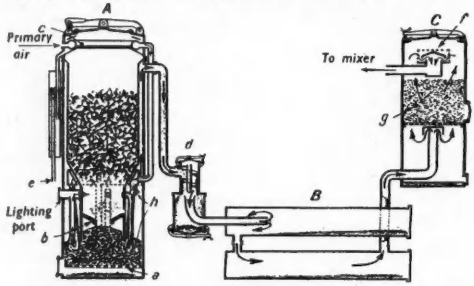


Fig. 2. **Standard Design No. 2 for Wood Fuel.**
(*A*) Producer, (*B*) Cooler-De-duster, (*C*) Filter.
(a) Grate, (b) Removable cone, (c) Condenser, (d) Centrifugal dust catcher, (e) Condensate drain, (f) Safety filter, (g) Granulated cork, (h) Air preheating part.

arrangement comprises the producer proper, a centrifugal gas de-duster and a cleaning system. The shape and construction of the furnace are visible in Fig. 2. In this design the hopper is double-walled for preheating of the fuel by the gas leaving at the top. From the gas producer the gases are passed to the tangential inlet of the dust collector *d* (Fig. 2) which is equipped with a helical spiral to impart the required whirling motion to the gas. Collection of the separated dust is seen to be effected in the cylindrical base of the dust catcher. Further gas-deducting takes place in the combined cooler and dust collector composed of two cylindrical elements arranged in series. The scrubber is a vertical cylinder divided into two compartments by means of a horizontal perforated sheet metal diaphragm carrying the scrubber filling *g* composed of granulated cork.

Standard model No. 3 is given in Fig. 3, which

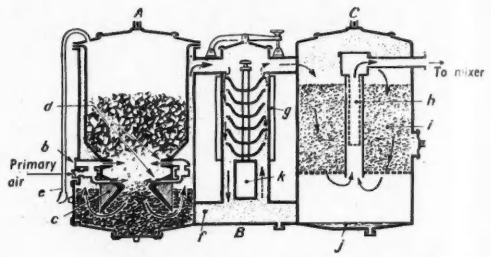


Fig. 3. **Standard Design No. 3 for Wood Fuel.**
(*A*) Producer, (*B*) Two coolers, (*C*) Filter.
(b) Lighting tuyere, (c) Refractory cement, (d) Two staggered rows of tuyeres, (e) Vent, (f) Bottom collector of cooler, (g) Cooling fins, (h) Safety filter, (i) Cork, (j) Condensate, (k) Handwheel opening

shows the principal details of the installation. Here the furnace proper consists of a ring-like recessed piece of high temperature resistant metal affixed to the lower part of the hopper. Primary air is admitted at the upper level of the furnace periphery at the point where the fuel enters the furnace ring, while secondary air is admitted through nozzles arranged peripherally at the bottom part of the furnace ring. Owing to the jacketed construction of the hopper part, the ascending gas stream in the jacket serves to preheat the hopper contents.

The gas cooler consists of two vertical cylinders with vertical ribs, their lower gas outlets being connected to a common chamber serving both as gas collector and dust container. A number of bowl-like buffle plates are seen to be arranged in the cooler proper. The scrubber of the down-flow type is of cylindrical shape and contains a bed of granulated cork. The condensate is collected in the bottom of the scrubber.

The design representing standard model No. 4 is outlined in Fig. 4. The body of the gas producer is seen to be a jacketed design, the lower part of the jacket being passed by the leaving gas, while in the upper part the hopper wall is perforated so that the jacket space can act as condenser for the volatile gases. This distillate is drained off through the line *b*. The furnace is entirely of metal construction, the lower truncated part being removable. The air is preheated in the annular jacket surrounding the furnace section, and subsequently enters the furnace through a number of tuyeres, peripherally arranged. The furnace bottom is formed by a rotary grate equipped for external operation.

The scrubber is comprised of three stages through which the gas passes in succession. The first stage serving for de-dusting of the gas is followed by a washer-scrubber stage, and in the third stage the gas is filtered through a layer of cork. Final cleaning of the gas is carried out in the final filter composed of a number of filter cartridges arranged in parallel.

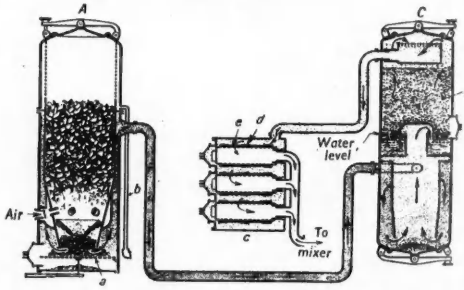


Fig. 4. **Standard Design No. 4 for Wood Fuel.**
(*A*) Producer, (*C*) Filter.
(a) Rotary grate, (b) Condensate drain, (c) Final filter of cartridge type, (d) Supporting straps, (e) Filter sheet, (f) Cork.

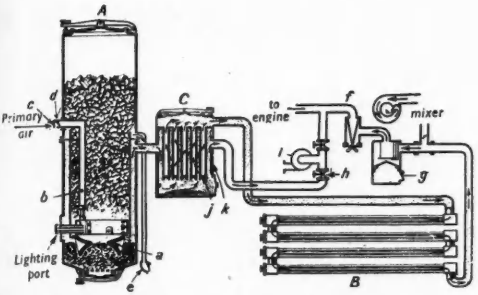


Fig. 5. **Standard Producer Gas Installation No. 5.**
(*A*) Producer, (*B*) Cooler, (*C*) Filter.
(a) Cast-iron cone, (b) Air supply to tuyeres, (c) Air supply damper, (d) Check valve, (e) Condensate drain-off, (f) Safety filter, (g) Centrifugal separator, (h) Valve (open during starting), (i) Starting blower, (j) Ceramic filter cylinders, (k) Glass silk insulation.

The standardised model No. 5 is illustrated in Fig. 5. The furnace itself is contained in a cylindrical inset containing the tuyere ring which may be of iron plate or of cast iron; while the cone arranged below the tuyere ring is made of a special grade of cast iron. The tuyeres proper are arranged in two rows with independent air supply, each tuyere row having its individual air control valve. As indicated in Fig. 5, the air supply lines are equipped with a check valve.

The gas cleaner consists of a number of ceramic filters in cylinder form held between two tube sheets. The flow of the gas through the filter bodies is directed radially inward so that the impurities in the gas are deposited on the outer filter surface. According to space conditions, the filter casing can be arranged vertically (as shown) or horizontally. Having deposited ts dust contents in the filter, the gas passes on to the cooler *B*, in which the condensable gases are retained in liquid form. Any moisture left in the gas after its passage through the cooler is subsequently separated out in the cyclone type dryer and mixer *g*.

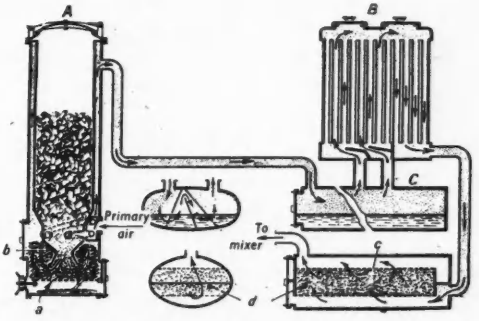


Fig. 6. **Standard Installation No. 6 for Wood Fuel.**
(*A*) Producer, (*B*) Cooler, (*C*) Preliminary Filter.
(a) Movable grate, (b) Furnace cone, (c) Filter, (d) Cork.

Standard design No. 6 is the Imbert producer shown in Fig. 6. The body of the producer is double-walled, with the outgoing gas passing upward through the annular space between the walls. The lower part of the body is equipped with three doors, one serving for removal of the ash and the two others for stoking and inspection. The grate at the bottom is hand-operated by means of an external lever. Inner and outer parts of the body are flanged together at the top as shown in Fig. 6, where the disposition of the air tuyeres is likewise indicated. The first-stage cleaner is a washing device of elliptical shape with three baffles so arranged that the gas makes two passes through the washing water. The cooler arranged above the washing device is of the tubular type, the vertical tubes extending between an upper and a lower tube sheet. Proper direction of the gas flow through the tubes is ensured by a set of baffles as shown. The final cleaning stage is contained in an elliptical vessel provided with a cork filter bed *c*, the gases passing through the filter in upward direction.

TESTING OF MOTOR AND GENERATOR ARMATURES BY MEANS OF CATHODE RAY TUBES.

By F. F. Molero. (From *Revista Electrotecnica*, Buenos Aires, Vol. 29, No. 6, June, 1943, pp. 301-304.)

In addition to the well-known uses of cathode ray tubes, there are a large number of applications which are less well known. Thus, for instance, cathode ray tubes can be used in a simple manner for the testing of the armature of small motors and dynamos, such as are used for starting and lighting purposes in automobiles, or for fans, dust exhausters, sewing machines, washing machines, and other household implements.

This testing is as useful to the manufacturer of these machines as it is to repair shops in the investigation of faults. Generally, cathode ray oscillographs may well prove to be an important testing device, largely excelling any other methods and devices hitherto used.

Both in the winding and in the rewinding of armatures, and also in the invesigation of faults on the running machine, any existing short circuits between the commutator segments or ground faults can be observed by this means.

The procedure followed in the testing of the armatures of small motors and dynamos of this kind consists simply in verifying the resistance of each coil, or in measuring its insulation with respect to the motor shaft or the core. A commonly employed test circuit is shown in Fig. 1. Owing to the inertia of the ordinary galvanometer used with this circuit, testing of the individual windings of the armature can only be effected by slowly rotating the rotor under the brushes in order to be able to perceive any faults that may exist in any one of the coils. A considerably greater testing speed—particularly important in mass production—can be achieved with the use of a cathode ray tube; and this also permits the use of alternating mains current. A set-up of this kind for testing armatures of high resistance is shown in Fig. 2, where the brushes are so

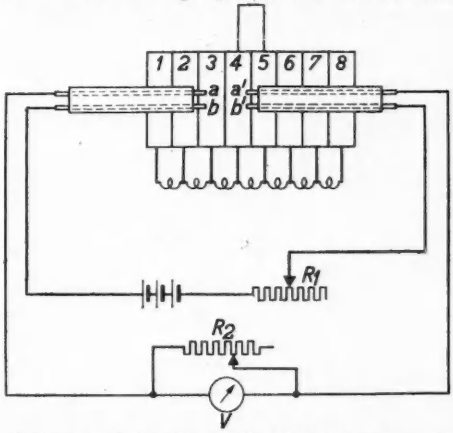


Fig. 1. Commonly employed method for testing armature coils.

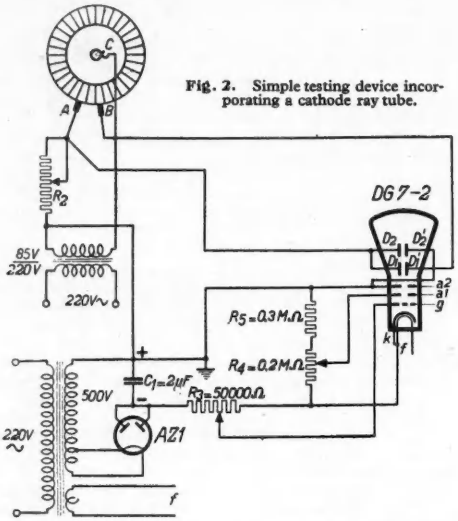


Fig. 2. Simple testing device incorporating a cathode ray tube.

positioned that one coil of the armature always lies between the brushes. By means of the cathode ray tube, it is possible to determine the difference in the electrical resistance of a normal and of a faulty coil. For this purpose the brushes are connected to a resistor of approximately 600 ohm.

This resistor is so adjusted that under normal conditions its resistance equals that of the coil under the brushes. As the resistor and the coil represent a potentiometer in parallel with the secondary windings of the transformer (Fig. 3), the screen of the tube will show a sinusoid forming an angle of 45° with the vertical. In Fig. 3, R_1 represents the resistance of the coil, and R_2 that of the resistor. If the armature is rotated—thus connecting one coil at a time—a line r-s will appear on the screen; and with quick rotation both the lines p-q and r-s will be visible. In case of a short circuit between the segments of the commutator, the resistance R_1 will be zero and a vertical line will appear on the screen. In case of a defective soldered joint, R_1 will be large in comparison to R_2, so that practically the full transformer voltage will exist at the terminals of R_1. This will result in the appearance of a horizontal line on the screen. Similarly, a ground fault, making itself felt over the connection C (Fig. 2) will appear as a vertical line on the screen. In view of

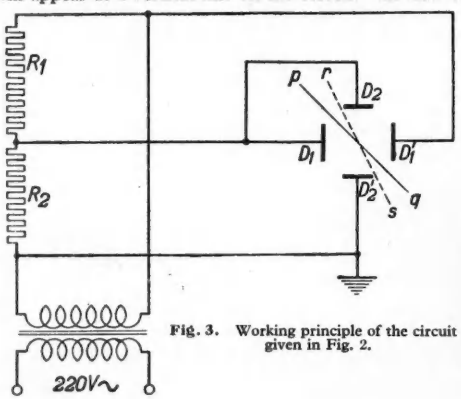


Fig. 3. Working principle of the circuit given in Fig. 2.

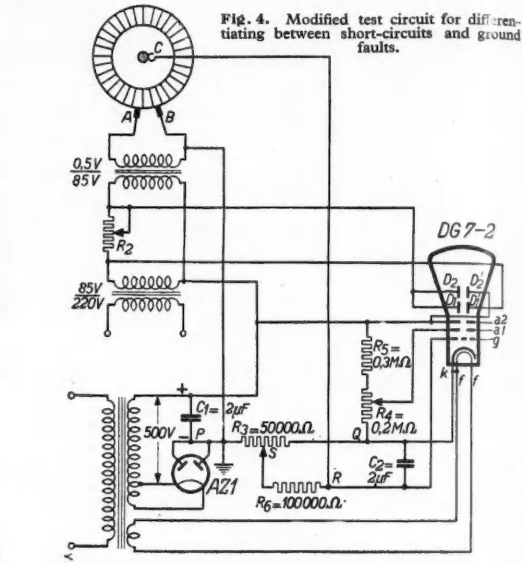
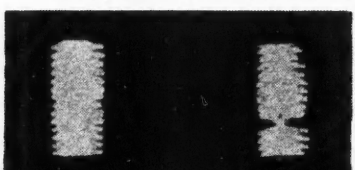


Fig. 4. Modified test circuit for differentiating between short-circuits and ground faults.

the fact that a certain voltage is required to produce a line on the screen of the tube, the transformer voltage must be chosen in accordance with the magnitude of the resistance of the armature coils.

In order to minimise the resistance of the connecting wires, the testing transformer should be placed as near to the brushes as possible. Where it is desired to differentiate between an interior short circuit of the coil and a ground fault, the circuit shown in Fig. 4 may be employed. Here the brush B is connected to point P, while the grid of the cathode ray tube is connected to the brush C contacting the shaft of the machine. The negative grid bias of the tube is obtained from the potential difference at the terminals of the resistor R_3 of 50,000 ohms.

Point Q is electro-positive with respect to S, which is the sliding contact of the potentiometer R_3. The latter serves to adjust the light intensity of the line appearing on the screen. Point P must be electro-negative with respect to A to the extent necessary to effect complete blocking of the cathode ray. If a short circuit is produced between the contact point C and the brush B, the points R and P will likewise be in short circuit, and the grid assumes a negative potential resulting in complete disappearance of the ray. A potential difference of 30 V. is highly suitable for testing the insulation value between the coil and the body of the motor. The condenser C_2 is provided for absorbing disturbing alternating potential differences which may occur between the grid and the cathode. By the various indications produced on the screen, this test circuit permits identification of the various types of faults that may occur, such as short-circuit between the brushes A and B, defective soldered joints on the commutator, broken coil wires, and faulty insulation. An actual image, as it is seen on the screen of the tube, is reproduced in Fig. 5, where the existence of a short circuited coil is clearly indicated in the right-hand oscillogram.



a b

Fig. 5. Reproduction of actual oscillogram (*a*) Normal, (*b*) One coil short-circuited.

CHARACTERISTICS of BLACK COATINGS for NON-FERROUS METALS

By WALTER R. MEYER, Technical Director, Enthone Company. (From *Product Engineering*, Vol. 14, No. 12, December, 1943, pp. 792-794).

NEW blackening processes for non-ferrous metals which are glare-proof, able to withstand corrosive elements in service and have good wear resistive qualities are discussed.

They comprise four types :—oxide, metal complexes, sulphide, and metallic. Black oxide coatings are found superior to the other three types and have excellent adhesion, good corrosion protection, temperature stability, and are a good base for organic finishes. This class of coating can be sub-divided into methods of application—viz., direct chemical oxidation at low or high temperature, electrolytic oxidation, anodic deposition, chemical reduction from a higher valency and chemical precipitation.

Direct chemical oxidation of copper produces the most satisfactory black finish of any method. It utilises aqueous solutions of strong oxidising agents in the 150-250 deg. F. temperature range. This finish is widely used on marine equipment made of brass, copper and bronze because of its salt-spray resistance.

Cupric-oxide coatings have heat radiation and heat stability characteristics of definite advantage for heat-radiating apparatus. The finish also is used on certain high-purity copper parts of electronic power tubes and makes an excellent base for finishing copper alloys with paints, enamels and laquers. Before development of this finish, there was no entirely satisfactory method of treating copper alloys for painting.

Any metal capable of being copper plated can be given a cupric-oxide finish by copper plating and oxidizing. This technique is rapidly replacing black-nickel plating because of the much greater ease of the process over black nickel and because of the greater depth of colour and adhesion of the cupric-oxide finishes. Cupric-oxide finishes cannot be satisfactorily applied directly on alloys with less than 65 per cent copper. However, there is no other direct method for blackening such alloys and a plating operation is, therefore, necessary.

An interesting method of blackening nickel has been described by B. B. Knapp in a patent assigned to the International Nickel Co. (U.S. Patent No. 2, 221, 641, Nov. 12, 1940). The method requires the use of an oxidizing agent, such as hydrogen peroxide or ammonium persulphate, together with a substance that forms small amounts of sulphide ions in solution, such as ammonium thiocyanate, and the bath is operated in a low pH range from 1 to 2.

Direct oxidation at high temperatures is one of the oldest methods for producing black-oxide coatings. When copper is heated in air or oxygen up to 800 deg. F. and over, heavy coatings are formed consisting of an outer layer of smooth, black cupric oxide and an inner layer of red cuprous oxide (U. R. Evans, " Metallic Corrosion, Passivity and Protection," Edward Arnold & Co., 1937, p. 98). The method, however, is not used frequently for blackening copper.

Electrolytic oxidation of aluminium and zinc has grown in popularity because of the war. Aluminium and its alloys can be oxidised by anodic treatment in sulphuric acid, chromic acid, and other electrolytes and the coatings thus formed can be dyed and sealed. Usually sulphuric acid solutions are used because heavy oxide coatings of the order of 0.0001 in.-0.003 in. are formed and can be dyed easily in almost any colour.

A method observed in Germany before the war involved oxidation of aluminium. The oxide was impregnated with a silver emulsion and the surface was then sensitized, using it in the manner of photographic print paper. Any pattern could be reproduced upon the aluminium. This method may reach considerable importance after the war when aluminium is freely available.

Zinc-base die castings can be darkened by making them the anode in a dilute sodium hydroxide solution. Zinc and zinc alloys are blackened by molybdate solutions and by nickel-ammonium thiocyanate solutions.

Chemical reduction from the higher valence state is highly successful commercially in the case of molybdenum sesquioxide coatings for blackening zinc and zinc alloys. Electroplated zinc, hot galvanised coatings, rolled zinc, and zinc-base die castings can all be blackened satisfactorily. The method involves proper cleaning of the work and placing it in buffered molybdate solutions operated from room temperature to the boiling point. The zinc reacts with the solution, reducing the molybdenum from the hexavalent state in solution to the sesquivalent state which forms as an adherent, smooth, black coating upon the surface. The coatings range in thickness from 0.00002 in. to 0.00005 in.

An electrolytic method is available commercially that co-deposits nickel and molybdenum oxide by electrolytic reduction from soluble molybdates. The adhesion of such deposits is not good on all basic metals but they adhere particularly well to zinc.

A method very widely used for depositing a blue-black oxide coating on brass is the so-called " copper carbonate-ammonia " method. The bath utilizes basic copper carbonate to which ammonia has been added to form a soluble blue complex. The process is good for depositing a blue-black colour on brasses containing from 65 to 80 per cent copper. Low-copper alloys do not react satisfactorily and high copper alloys give a grey-coloured deposit. Low-copper brasses, such as 60:40 brasses are best blackened by copper plating and then blackening the copper by direct oxidation.

The corrosion protection afforded by this type of coating is not good, being of the order of 2 hr. in the salt spray.

Metal Complexes.

Black-nickel coatings have been used for many years, particularly on typewriter parts. The coatings are deposited from solutions containing nickel, zinc, the thiocyanate radical and the ammonium radical. A typical analysis of the coating is sulphur, 8 per cent, NH_3, 4 per cent, carbon, 4.5 per cent, nickel, 23 per cent and zinc, 48 per cent.

The coatings are almost structureless and have only fair ductility. However, when the solutions are run properly thick deposits of the order of 0.001 in. can be formed. The corrosion protection offered is poor and protection must be obtained by an undercoating.

Black-nickel solutions are used to a limited extent for depositing a black coating on zinc surfaces without the use of electric current. Such solutions are not good for electroplated zinc because the zinc plate is appreciably dissolved during the treatment and there is formation of white basic zinc salt that adheres to the finish.

Sulphide Coatings.

Black sulphide coatings on non-ferrous metals are easy to apply. They are formed on copper and silver by immersion in dilute sulphide solutions, and offer little or no corrosion protection to the base copper and are not good bases for baked organic finishes. Also they are prone to " crystal spotting."

More stable and adherent black coatings are formed on silver and gold by heating and decomposing tellu-

rium chloride solutions which leave a deposit of tellurium and silver telluride on silver and probably just tellurium on gold.

A method of blackening aluminium by the use of selenium has been patented. It involves immersing the aluminium in a solution containing " . . . a mixture of selenium compound which produces selenous acid, and a soluble halogen salt of copper."

Metallic Black Coatings.

Smut-type black deposits can be formed on zinc, cadmium and iron by using acidified solution of arsenic or antimony. However, the deposits have poor adhesion and cohesion, and in the case of arsenic there is danger of poisonous arsine formation. Besides, arsenic-coated metal is a health hazard. In addition, the formation of such deposits on plated coatings results in attack on the coating by the acid on the blackening solution.

Chief drawback associated with these coatings is the high current density required namely above 1,000 amp. per sq. ft.

DOUBLE-EXPOSURE RADIOGRAPHY

By JAMES RIGBY. (From *Transaction of the American Society for Metals*, Vol. 31, No. 9, September, 1943, pp. 599-608).

To detect the vertical position of flaws in metal parts, it is possible to make two stereoscopic exposures on the same film. The vertical height above the film of the flaw producing the image may be found by the following formula (No. 1): No (flaw-film-distance) =

$$\frac{\text{focal dist.} \times \text{image shift}}{\text{tube shift} + \text{image shift}}$$

By placing a lead marker on the specimen being radiographed and noting the shift of its image on the film, either the flaw-film distance or the focal distance may be eliminated. By assuming the X-rays to be parallel a simplified and sufficiently accurate formula (No. 2) utilizing the lead marker is derived as follows: Flaw-film dist. =

$$\frac{\text{marker-film dist.} \times \text{flaw image shift}}{\text{marker image shift}}$$

Experiments were conducted to determine the accuracy with which holes could be located in aluminium and steel castings by these methods. The tube used was a Phillips Metalix Macro 200 p.k.v. type with water-cooled stationary anode. All exposures were made at 5 ma. The time was measured from an arbitrary position of a protective rheostat because the voltage had to be brought up slowly by means of the rheostat. The perpendicular focal distance (target to film) was taken as 91.8 cm.

Test parts for aluminium were unmachined cast aluminium bars, 48 × 7.5 cm., of different thicknesses. Each casting had twelve holes, 6.4 mm. in diam. of varying depths. The maximum possible error in recording these depths and the thickness of the casting is about ± 0.3 mm. Test castings 1 and 2 in. thick were radiographed with the holes on top and also next to the film. Those 1¼ and ¾ in. thick were superimposed and radiographed together with the holes at the inner faces. Non-Screen film with 0.2 mm. lead filters was used.

Steel Tests.

For steel, test specimens consisted of cast steel blocks 11 by 3.8 by 1.3 cm. and 11 by 318 by 215 cm., with grooves 1.3 cm. wide in which were fitted slotted bars of two kinds. The first contained slots 0.3 cm. wide and varied in depth from 0.25 to 2.0 mm.; possible error ± 0.04 mm. The other had slots 0.32 cm. wide, varying in depth from 0.5 to 3.0 mm.; possible error, ± 0.1 mm. Two of each size were radiographed together, one slotted bar on top and one between, making the total thicknesses 2.5 and 5.1 cm. High-speed film with two intensifying screens was used as Non-Screen film was unsatisfactory.

In working with aluminium it was found that drilled holes greater than 10 per cent of the casting thickness produced images of such clarity that their tops and bottoms could be located and the depths of the holes estimated. The errors in the calculations of depth were usually of the order of —50 per cent. The location of the holes themselves was found to be possible with errors usually much less than ± 10 per cent of the casting thickness and often under ± 3 per cent when sensitivity was under 4 per cent. As the experimental holes were of uniform shape, these errors correspond to optimum conditions. A tube shift of 1 ft. was found adequate except when the holes were near the bottom of the casting when a larger shift gave better results.

In working with steel, it was not found possible to distinguish between images of the tops and bottoms of the slots so each measurement was made from the edge of the image and recorded as the middle of the slot. Results of the experiments compared favourably with those on aluminium. The desirability of good technique, giving greater sensitivity was clearly shown. With both metals it was noted that the errors in using the second equation were more negative than those made with the first.

DEVELOPMENT OF CAST CRANKSHAFTS.

(From *Diesel Progress*, Vol. 9, No. 12, December, 1943, p. 60).

THE COOPER-BESSEMER CORPORATION has recently released information showing results obtained in exhaustive tests for replacing engine crankshafts of forged steel with cast meehanite crankshafts. A comparison of the methods and materials used to produce heavy-duty engine crankshafts shows the unusual savings that can be effected in modern casting procedure over conventional methods.

Just recently Cooper-Bessemer conducted its most spectacular test using a 6-throw shaft casting, 7 ft. 8 in. long, with 5⅜ in. pins and journals to replace the forged steel shaft in an 8¾ in. by 11 in. diesel engine. First, the engine was operated at 900 pounds peak pressure for 20,000,000 revolutions at 900 revolutions per minute. When the bearings were removed and whiting applied, no distress was apparent in the shaft. During the next phase of the test, 20,000,000 additional revolutions were run at 900 revolutions per minute and 1000 pounds peak pressure. Bearings were again removed, whiting was applied and the shaft was still in perfect condition. Engineers, in a determined effort to break down the engine, then operated it continuously in a peak critical test for six days. During the night of the sixth day, a change in operating efficiency indicated that the shaft had finally failed.

Although the engine was operated for 12 hours after all indications had pointed to a cracked shaft, a subsequent examination showed that this severe test, despite the additional running time, had caused a fracture extending only half way through the shaft and it still continued to function, attesting to the strength and high endurance of meehanite metal.

TECHNICAL NEWS

Announcements in this section include News relating to British Firms, and on Equipment produced by British Manufacturers.

Available literature may be secured by addressing a request to the advertising department of "The Engineers' Digest," or by writing direct to the manufacturer and mentioning "The Engineers' Digest" as a source.

- ✦ NEW EQUIPMENT
- ✦ BOOKS AND CATALOGUES
- ✦ BUSINESS CHANGES
- ✦ PERSONAL
- ✦ NOTES

BRITISH STANDARDS

(Copies of British Standards may be obtained from the British Standards Institution, 28, Victoria Street, London, S.W.1.)

DIAMOND TIPPED TURNING TOOLS (No. 1148:1943). THE British Standards Institution has now published a war emergency specification for diamond tipped turning tools, which has been prepared by co-operation between the British Standards Institution and the Tool Technical Panel of the Ministry of Supply, Diamond Die and Tool Control. Like the standard for boring tools it is primarily a dimensional one, and establishes the dimensions for two patterns of shank type turning tools, using respectively a "ball type" tool, permitting of universal adjustment to give side rake and either positive or negative top rake, and an "adaptor type" tool accommodating standard diamond tipped boring tools and providing either a 2° positive or a 2° negative rake. So that the fullest benefit may be obtained from the use of diamond tools, the specification includes some detailed introductory notes which have been prepared by the Diamond Die & Tool Control to give guidance in regard to such matters as the smooth running of lathe spindles, the foundations and mounting of lathes, and the setting of the tools. Price 2 -.

P.D. 160, P.D. 161, AND P.D. 162 AMEND. ALUMINIUM ALLOY SHEETS AND COILS, AND ALUMINIUM COATED ALUMINIUM ALLOY SHEETS AND COILS FOR AIRCRAFT PURPOSES.

AN Amendment Slip has just been issued to the three above British Standard Specifications in order to cover sheets and coils supplied in the as-rolled condition where bending or forming operations are necessary at the aircraft manufacturers works. The relaxations which are allowed for the material supplied for this purpose are set out in the amendment slip. The reference numbers of these slips are P.D. 160, 161 and 162 respectively, and copies are obtainable from the British Standards Institution, 28, Victoria Street, London, S.W.1. on application.

BS/STA 23. TERNEPLATE.

THE above specification just issued by the B.S.I. for the Ministry of Supply provides for terneplate of a quality suitable for use where soldering properties superior to those of ordinary terneplate are essential, for example, for the manufacture of amunition boxes. The requirements covered by the specification relate to the basis sheet and the quality and thickness of the tin coating.

The methods of testing are fully laid down. Price 6d. each (12 or more copies 3d. each).

REVISED STANDARD FOR SYNTHETIC-RESIN BONDED-PAPER SHEETS.

FOR some years there have been two separate specifications for synthetic-resin bonded-paper sheets for electrical purposes, namely B.S. 316 and B.S. 547, which have now been amalgamated and superseded by B.S. 1137, just issued by the British Standards Institution.

This new specification relates to sheets for use as electrical insulation at power frequencies, and separate proposals are in preparation for a revised specification for synthetic-resin bonded-paper tubes. In the meantime B.S. 316-1929 is still applicable to tubes.

The specification is based on technical data supplied by the Electrical Research Association, and with a view to reducing to a minimum the amount of testing required the specification has been divided into mandatory and optional clauses, the former being limited to the tests necessary for the rejection of inferior material. Price 2 -, post free.

B.S. NO. 1141. WAR EMERGENCY B.S. FOR SECONDARY ZINC ALLOY FOR DIE CASTING.

THIS specification just issued, has been prepared in order to make use of remelted redundant zinc alloy castings conforming to B.S. 1004. A large tolerance on copper has been allowed and the limits for other impurities, e.g., lead and tin have been raised above the low limits of B.S. 1004.

The initial physical properties of the alloy will vary between those of Alloy A and Alloy B, but it is urged that the application should be confined to uses where maintenance of physical properties and dimensional stability are not of primary consequence. Price 1s.

PERSONAL.

The Right Hon. Lord Brabazon of Tara, M.C., P.C., has been elected Chairman of Messrs. Tungum Sales Co. Ltd.

Mr. Peter Burt, J.P., M.I.E.E., the originator of the single-sleeve valve engine died recently at the age of 88. Mr. Burt was Chairman of Messrs. Acme Wringers Ltd.

Mr. E. Crowther has been elected Director of the Newcastle-upon-Tyne and Gateshead Gas Company.

Major-General H. E. Davidson, C.B., D.S.O., M.I.Mech.E., M.I.Mar.E., has been elected a Director of Messrs. Robert Stephenson & Hawthorns Ltd.

Capt. H. Leighton Davies, C.B.E., and Mr. P. B. Brown, have been nominated Vice-Presidents, and **Mr. W. B. Baxter** and **Mr. W. J. Dawson** have been elected Members of the Council of the Iron and Steel Institute.

Dr. Oscar Faber has been elected President of the Institution of Heating and Ventilating Engineers.

Mr. W. J. Hadfield, C.B.E., M.Inst.C.E., for many years city engineer and surveyor of Sheffield, has died on the 25th of January at the age of 72.

Mr. D. P. Oliver has been appointed Director of Mining Supplies, Ministry of Fuel and Power.

Mr. W. Findlay Seivewright has been appointed Manager of Publicity and Development of the International Meehanite Metal Co. Ltd.

Mr. J. Lockwood Taylor, D.Sc., has been appointed Chief Technical Officer of Messrs. King Aircraft Corporation Ltd.

Dr. A. C. West, B.Sc., A.M.I.Mech.E., M.I.Mar.E., has been appointed Director of Robert Gordon's Technical College, Aberdeen.

Mr. S. E. Whitehead has been appointed Managing Director of Southampton Gas Company.

Group-Captain F. Whittle, C.B.E., B.A., F.R.Ae.S., the inventor of the jet-propelled aeroplane, has been conferred upon the degree of M.A. by the University of Cambridge.

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INSTITUTION OF NAVAL ARCHITECTS.

THE eighty-fifth Annual Meeting of the Institution will take place on the 29th of March, 1944, at noon in the Library of the Royal Society of Arts, John Adam Street, Adelphi, London, W.C.2.

The Spring Meetings, 1944, will take place on April 19th, 20th and 21st in the Hall of the Institution of Mechanical Engineers, Storey's Gate, London, S.W.1. The proceedings will commence each day at 2.30 p.m.

In addition a **Boiler Symposium** will be held on May 10th and 11th in the Hall of the Institution of Mechanical Engineers for reading and discussing a group of papers on the application of water-tube boilers to merchant ships.

ANGLO-POLISH AUTOMOBILE DISCUSSION.

ON January 16th an informal discussion took place jointly under the auspices of the Institution of Automobile Engineers and the Motorization Section of the Association of Polish Engineers in Great Britain. Mr. Geo. Lanchester, President of the Institution, was in the chair and introduced the authors of the following short papers—(1) *The Motor Car and General Engineering Industry in Poland*, by Jan Dabrowsky, M.I.Mech.E., (2) *Types of Motor Vehicles Suitable for Polish Road Conditions*, by T. Z. Marek, A.M.I.A.E. Copies of the papers may be obtained from the Institution of Automobile Engineers, 12 Hobart Place, London, S.W.1.

The Institution of Mechanical Engineers announces that, thanks to arrangements made by the War Organisation of the British Red Cross and the Order of St. John of Jerusalem, no less than 37 candidates recently sat for the Institution's examinations in Prisoner-of-War Camps in Germany. No fewer than 34 or 92 per cent passed with an exceptionally high average percentage marking. The results reflect credit both on the candidates and on members of the Institution and others in the Camps who organized classes and acted as instructors.

ACTIVITIES OF C. A. PARSONS & COMPANY LTD.

MESSRS. C. A. PARSONS & CO. LTD., Heaton Works, Newcastle-upon-Tyne, have sent us the following details of their present activities :

During the past twelve months many important orders have been received for turbo-alternators required for both this country and abroad. These include a 30,000 kW. turbo-alternator for a power station in Australia, the condensing plant and feed heating system being manufactured in that country to our design and drawings. The turbine will be of the two cylinder tandem type with double flow low pressure cylinder, and will operate with steam at a pressure of 585 lb. per sq. in. at the top valve, a temperature of 800° F., and an exhaust vacuum of 28.75 in. Hg. Four-stage feed water heating by steam extracted from the turbine will be employed. The alternator will generate at 11,000 volts.

A 40,000 kW. tandem turbine to operate with steam at a pressure of 615 lb. per sq. in. at the stop valve, a temperature of 815° F., and an exhaust vacuum of 28.5 in. Hg. is on order for a power station in South Africa. This turbine will be arranged also for four-stage feed water heating. A 25,000 kW. and a 12,000 kW. turbo-alternator, both arranged for passing out steam after partial expansion for process purposes, are also on order for export. Amongst the smaller machines on order are a 2,500 kW. turbo-alternator and a 2,500 kW. "pass-out" turbine for industrial purposes.

Another interesting order in hand is for the supply of the propulsion machinery for three turbo-electric ships, including the turbo-alternators, driving motors and auxiliary alternators. The turbines will consist of two cylinders in tandem and will be designed for operating with steam at a pressure of 800 lb. per sq. in. and 850° F. The steam will be reheated after expansion in the high pressure cylinder to 850° F. before entering the low pressure cylinder. The turbo-alternators are of standard construction and each is connected through a hand operated control cubicle to a double unit motor which drives a single screw. Although the double unit motor has a single housing, each half is so constructed that the electrical circuits are independent of one another. An auxiliary supply is afforded by a diesel driven alternator which can be connected to either half of the motor for emergency use.

The largest machine set to work in this country during the period was a turbo-alternator of over 50,000 kW. capacity to operate with steam at a pressure of 650 lb. per sq. in. and a temperature of 825° F.

Overseas we have completed turbo-alternators for outputs of 16,000 kW., 7,500 kW., 6,250 kW. and 2,500 kW. The steam conditions for these sets vary between a pressure of 250 lb. and 450 lb. per sq. in. and a temperature of 570° F. and 750° F. The second turbo-geared centrifugal pump for the Rand Water Board, South Africa, has been completed and is in service.

The Transformer Department has been occupied mainly with manufacturing large numbers of small and medium sized distribution transformers. Among the major contracts completed are two 20,000 kVA., 5,750/20,000v. O.F.B., power station step up transformers, a second 2,200 kVA. shell type furnace transformer delivering 29,333 amperes, and seven 3,333 kVA. 110/6.35 kV. single phase transformers (forming two three phase banks with one spare) for a British Dominion. A duplicate order for a three phase current limiting reactor, giving 10 per cent reactance in a 22,000 kVA., 6,300 volt system, heads a long list of reactors completed during the year for both home and export markets.

Among the more important orders received are four transformers forming a 31,500 kVA., 115/10.5 kV., three phase bank with one spare unit, and a 10,000 kVA., 38.5/6.6 kV., three phase transformer all for export. Home orders received during the year consist mainly of distribution, welding and furnace transformers.

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